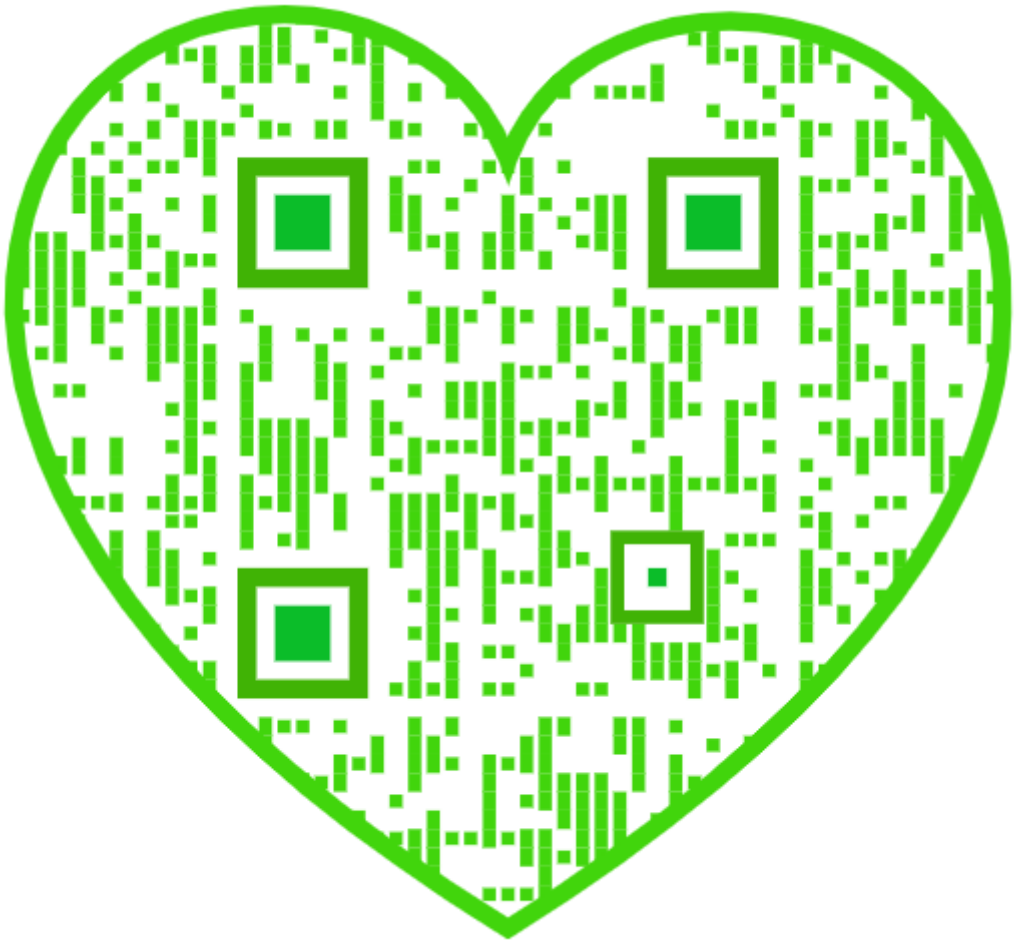

TREATISE

ON

MODERN HOROLOGY.

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A TREATISE

ON

MODERN HOROLOGY

IN

THEORY AND PRACTICE

TRANSLATED FROM THE FRENCH OF

CLAUDIUS SAUNIER

EX-DIRECTOR OF THE SCHOOL OF HOROLOGY AT MACON

BY

JULIEN TRIPPLIN, F.R.A.S.

BESANÇON WATCH MANUFACTURER

AND

EDWARD RIGG, M.A.

ASSAYER IN THE ROYAL MINT

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This Work was honoured with a First Class Medal in 1867, *and a Gold Medal and Honourable Mention in* 1868;
and the Translation by a Diploma of Merit, Melbourne, 1880, *and a Gold Medal, Paris,* 1881,
"*For Services rendered to the Cause of Horology.*"
The Translators have also received the Palmes of the French Academy, 1885, *and the thanks of the British Horological Institute.*"

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AUTHOR'S PREFACE.

THE publication of the first edition of this work commenced in numbers, in the year 1861. I consider it necessary to mention this fact, because several of the original explanations it contains were, during its publication, given in the works of other authors without any indication of the source from which they had been derived.

Technical works, especially such as treat of the watchmaking art, would without question be more easily understood if they could be made brief and didactic; but such a practice at the present day, when theoretical instruction is unfortunately but partially accessible to watchmakers, would defeat the object of the author, which clearly should be to place such information, as well as practical results thoroughly established by experiment, within the reach of manufacturers and all others interested in the subject.

This fact will not only explain the elaborate details which I have given in considering certain subjects, but will also justify both the plan of the work and the method adopted in the theoretical and practical explanations.

It is unnecessary to enumerate the novel features in the work, or to refer to the toil which its preparation has involved. Any one will be able, after consulting former treatises, to do me justice in this matter. I would only add that my chief aim has been to make the volume useful to the greatest possible number of those who live by our delicate and difficult industry,

and, at the same time, to be honest both in style and in the opinions I express; only asking that those who succeed me or who discuss the principles which I have laid down, will judge me as I consider myself at liberty to judge others.

I have pursued my researches and my various experiments just as they are here described; without prejudice, and endeavouring to be just and truthful in all cases, especially where I was compelled to discuss individuals.

I will conclude by the expression of a hope:

May a highly intelligent young student of the subject, with an honest heart, untainted by any of the paltry jealousies with which he may, as I have done, come in contact in the course of his labour, enter on the rough but useful path which I have opened up; may he surpass me in his success and I shall be the first to congratulate him, on condition that he is careful to respect names as well as opinions, which, although possibly at times erroneous, are nevertheless conscientious. He will thus do himself honour, and the practice of kindliness and modesty will become easy to him if he keeps in mind that sentiment of Pascal's, "We should see no farther than those who have gone before us, did not their knowledge serve as a stepping stone to our own."

CLAUDIUS SAUNIER.

TRANSLATORS' PREFACE.

VERY little explanation is necessary as to the circumstances that have led us to undertake the preparation of an English edition of Saunier's celebrated *Traité d'Horlogerie Moderne*. England has so long held a foremost place in the manufacture of instruments for the accurate measurement of time that the almost total absence of horological literature is a matter of no little surprise. Periodicals, such as that published by the British Horological Institute, are of great value as affording a medium for the interchange of ideas between watchmakers and others, and for discussing doubtful points in connection with their art, but such journals are only in a position to supply fragmentary instruction for the younger members of the craft, for whose education a systematic treatise, such as that of M. Saunier, is essential. Indeed, the technical ignorance, too often displayed by working watch and clock makers and jobbers, cannot be wondered at so long as this want is left unsupplied.

But the usefulness of such a treatise does not stop here. The manufacturer and the foreman will find in it a collection of technical details that cannot fail to be of the greatest service to them whatever be the branch of horology on which they are engaged, and the amateur will welcome the scientific discussion of an important and fascinating mechanical art in an exhaustive manner never before attempted in this country.

In all the mechanical industries great efforts are being made at the present day in the cause of technical education, but in order that such efforts may be successful it is of the first importance that sound textbooks and competent teachers should be provided. We believe the present volume well supplies the first of these wants as regards the science and art of the

subject with which it deals, and, although the number of those that are both able and willing to undertake the duties of instructors may, at present, be somewhat limited, it is reasonable to hope that, as the real demand for technical training increases, more teachers may be ready to help in the work.

When this translation was first undertaken, in 1877, we anticipated that a supplementary volume then in preparation by M. Saunier would be published in time to form part of the work. Although this hope has not been fulfilled we have arranged with the author to translate the appendix immediately on its publication. It will be devoted to the subjects of electrical and turret clocks, springing and timing and depths, as well as all the more recent researches connected with the art, and will be illustrated by 9 additional plates.

In the meanwhile we are preparing a *Watchmaker's Handbook*, based on the *Guide-manuel de l'horloger et Recueil des procédés pratiques* of M. Saunier, which, in addition to a detailed description of the various tools and instruments employed, will contain unusually full practical explanations of the mode of preparing and working metals, methods of constructing and repairing the several parts of watches, etc., together with many useful tables and receipts. To this work reference is frequently made in the present volume.

Feeling that the utility as a work of reference of such a treatise as that now presented to the horological reader depends in no small degree on the fulness of the index, we have prepared an entirely new and very full one containing over two thousand references; and it is believed that by referring to it, to the table of contents or to the key to the plates, the reader will in all cases be enabled at once to find the subject of his search.

Measures of length and weight have for the most part been given both on the imperial and metric systems, but tables are also given at the end of the volume by the aid of which any required conversion can be effected.

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PART I.

ESCAPEMENTS.

INTRODUCTION

TO THE STUDY OF ESCAPEMENTS.

Preliminary Considerations.

1.—Every machine intended for the measurement of time is composed of two distinct portions: One, the *train*, consists of a succession of toothed wheels, whose function it is to transmit to a definite point a motive force produced by a weight or spring; and the other, called the *escapement*, is a special mechanical appliance adapted to the end of a train of wheels in order to prevent a too rapid motion, and to *regulate* the expenditure of the motive force in such a manner that it is allowed to exhaust itself with the requisite slowness and uniformity.

The theory of depths should thus precede the theory of escapements. We shall, however, for reasons given in a subsequent paragraph (**1029**), commence with the latter; but it should be observed that a knowledge of these two theories, depending as they do on identical mechanical principles, is of equal importance for attaining to precision and a permanent adjustment of watches, clocks, and other horological appliances.

2.—Few branches of Mechanics have afforded a wider field for the ingenuity and sagacity of inventors than escapements; and there is hardly a single watchmaker of note who has not produced one or more of original design. Thus the number of escapements is known to exceed a hundred; but they do not constitute an *embarras de richesses*, for only from ten to fifteen of this number have been retained in use, and the others, as Moinet wisely remarks, are rather examples of what is to be avoided than patterns to be imitated.

Such a result was, however, only natural; the majority of those who have produced novel escapements have been nearly, if not quite, wanting in that mechanical knowledge which is indispensably necessary. In their vanity, as skilled workmen, they were satisfied that skill alone, the genius of natural inven-

tiveness, or rather some lucky and fortuitous hit, would enable them to discover the secret of the exact measure of time far better than a careful study of the laws of Mechanics. A mass of designs, false in principle, and more or less identical, was the consequence of this impression; indeed, in several instances, successive generations of inventors have followed up the application of one ingenious but erroneous idea, and kept dropping into the false track already trodden by their ancestors.

3.—It is a fact well worthy of note, which we especially commend to the notice of students, that all those escapements which have held their place, *were designed by men who were good mechanicians*, that is, men who supplemented a long and painstaking experience by sound theoretical knowledge.

Classification of Escapements.

4.—Escapements are commonly divided into three principal classes, which may again be considerably subdivided:

I. *Recoil* escapements, so named because, at a certain period of its action, the wheel moves backwards or recoils in a manner more or less marked. The verge escapement in watches and certain forms of anchor in clocks may be referred to as examples.

II. *Dead beat* escapements, characterised by the fact that, except during the actual impulsion, the wheel remains stationary, a point being supported either against the axis of the balance itself or against an accessory piece, concentric with this axis, which catches it in its movement of rotation. Such are the cylinder and duplex escapements in watches, and the pin escapement and Graham's escapement in clocks.

III. Lastly, *Detached* escapements, which are also dead beat escapements, but whose principal characteristic consists in the fact that the balance performs its vibration in absolute independence of the wheel, except during the very brief periods of impulsion and unlocking. The wheel, then, does not rest on the axis of the balance, but on an intermediate and distinct piece. The lever escapement in watches, the detent escapement in chronometers, as well as several forms of escapement employed in clocks, come under this category.

5.—Some authors introduce a fourth class, including the constant force or *remontoir* escapements, but both theory and experience go to prove that, except in a few special cases, these arrangements, so expensive and so difficult to construct and to

keep in repair, have objections when employed in time-keepers of small dimensions with no sufficient counteracting advantage.

It will be subsequently shown, especially when we come to the portion of the work relating to turret clocks, in what special cases remontoirs can render real service.

Incompleteness of this Classification.

6.—The division into recoil and dead beat escapements, although usual, is not sufficiently precise.

In the case of the duplex escapement, justly placed in the latter class, the wheel at a certain period moves slightly forward and then recoils to the same extent. Certain forms of anchor used in clocks produce a recoil, which, when the oscillations are short, commences or terminates an interval of rest. They thus partake, although in unequal proportions, of the nature of each class, and the same may be said of those which rest on one side and recoil on the other, etc.

Comparison between Recoil and Dead Beat Escapements.

7.—The majority of authors consider the recoil escapement, from the very fact of a recoil existing, to be the least perfect of those in ordinary use; but this decision is too absolute to be regarded as authoritative. The characteristics of the two classes of escapements are so manifestly different, that what is true of one, is not to an equal extent true of the other. The verge escapement, having a very considerable recoil, is justly placed lowest in the scale of comparison, and yet the duplex, notwithstanding a slight but visible recoil, surpasses some dead beat escapements in maintaining the going of a watch uniform.

8.—The recoil is sometimes taken advantage of to neutralise the inequalities of the motive power, as is shown by the frequent employment of small anchors in ordinary timepieces; but care is necessary, in such cases, to avoid all prejudicial friction, which would inevitably occasion variations. The regularity of clock escapements with a considerable recoil is, in consequence of this fact, inferior, in nearly all cases, to that which we should obtain by replacing them by dead beat escapements.

The expression "considerable recoil" is here used advisedly; for if the effect of the recoil is retained within definite limits, and if its action commences only at a particular period in the interval of rest, beneficial results are obtained. This, at least, has

been established both from theoretical considerations and from an experience extending over more than twenty years, by a skilful French mechanician, M. Rozé.

Comparison between Dead Beat and Detached Escapements.

9.—The escapements in which friction occurs nullify inequalities in the motive power sufficiently for ordinary purposes without being supplemented by the principle of isochronism, or by compensation for varying temperatures. But when the surfaces resting in contact are considerably reduced, these escapements lose their correcting properties; and if, on the contrary, they exceed certain limits, they give rise to the numerous perturbing causes occasioned by excessive friction, etc. Theory and experience have pointed out the best intermediate form, and it appears that the principal condition of attaining it consists in establishing a certain relation between the size and weight of the balance, and the radii of curvature of the surfaces between which friction takes place. This subject will be subsequently reverted to.

10.—The detached escapements employed in watches are considered preferable to those in which the rest is frictional, and a greater degree of precision can be attained by means of them. Their oscillations, except during a very brief interval, take place without contact with the motive power, and thus the friction is reduced to a minimum. It is, however, important to distinguish between the requirements of every-day life and those of science, for, as the mutual actions are more delicate, the balance more free, and the friction less, so the adjustment of the balance for temperature and the isochronism of the balance-spring must be more perfect. Thus, for example, the chronometer escapement, which gives such excellent results when these two influences are properly co-ordinated, gives, without their adjustment, less satisfaction to its possessor than a good lever movement; in this, nevertheless, the frictions are far more numerous, and it does not require, in the same degree, these two accessories in order to give results that are, after all, more than sufficient for ordinary purposes.

11.—In timepieces, the dead beat escapement is usually preferred to the detached escapement; the reason for this will subsequently appear.

12.—In conclusion, then, before asserting the excellence of any one form of escapement, or its superiority over any other, it is important first to clearly explain the use to be made of it.

GENERAL PRINCIPLES

Deduced from the Laws of Mechanics and from Observation.

13.—In order to judge of the merits of an escapement, whether old or novel, to foretell its good and bad qualities, to improve it or alter it by suitable modifications, and not at the same time to fall into old and disused forms, it is necessary to bring to bear on the study of the subject not only a rich store of observation derived from a systematic examination of old designs, and a long and sustained handicraft, but also a thorough knowledge of the general physical and mechanical conditions which are involved. Those who desire to grasp the question cannot afford to be ignorant of these points, still less those who hope to design something which, if not good, is at any rate novel. This ignorance is nevertheless sadly common amongst the inventing class.

A theoretical principle is a reliable guide, but, to apply it intelligently, experimental data are essential. The mathematical formula, in its unyielding strictness and its absolute precision, shows us the goal to be aimed at; but the practical man must choose for himself the route by which to reach it, for he alone is able to understand the difficulties involved, detecting the obstacles which either the nature of the material operated upon or the lack of mechanical method throws in his way.

We proceed to epitomise and to briefly develope the general principles which should guide a maker or inventor in designing an escapement. Our readers, whom we assume to be familiar with the practice of watchmaking, will see, after a careful perusal, how easy it is to apply these principles with advantage to the consideration of escapements now in use, and which will be described subsequently.

ELEMENTS OF APPLIED MECHANICS
For the use of Watchmakers.

14.—Not unfrequently at the present day one meets with mechanicians, even among the simple artisans, who possess a certain amount of theoretical knowledge, and know how to render it serviceable. Watchmakers, with but few exceptions, are very far from being favourably placed as regards pro-

fessional instruction, and this disadvantage is doubtless one of the causes which have contributed to bring about the present depressed state of Horology, taken as a whole.

The aim of the following work, which only requires *a little attention and natural intelligence*, accessible to everyone, for its proper understanding, is to remedy this deplorable state of things, by enabling each watchmaker, by himself, to acquire a clear notion of the laws of Mechanics, and to be able, through knowing the causes, to rectify many faults which pass current among the so-called enlightened public. He will derive this double advantage; he will rise in the estimation of his customers, and will be helped in his daily work in a manner he could not have suspected.

But does this imply that the following epitome pretends to be a treatise on Mechanics? No. Its intention is less ambitious, for it is written expressly for one particular class of readers. To take the practical man in hand, to lead him from facts which require but simple argument to those of a class somewhat more advanced; to instil into him healthy ideas, by impressing him, so to speak, with a desire for knowledge and a conviction of its use, at the same time smoothing down the early difficulties which lead to it: this is our aim, this our ambition.

Forces.

Of two kinds.—Mechanical Effect.—Power.—Resistance.

15.—Every cause which produces, modifies, or stops the motion of a body, is a force. It is either a *power* or a *resistance*.

Forces have two characters: the one active, the other passive.

They can produce a motion or else stop one already in existence, by acting in a direction opposite to it; this is the characteristic of *active forces;* such, for example, as the action of a weight, the effect of a coiled spring, the fall of a liquid, etc.

Passive forces can wholly or partially destroy motion, but are unable to produce it; such are friction, stiffness of cords, etc.

16.—It follows from these definitions that certain forces are active or passive, according to circumstances; thus we shall see that inertia which, in a moving body, is an active force, becomes passive when the body has been brought to a state of rest. The same is the case with the air, which may behave either as a power or a resistance.

Although the nature of force is unknown to us, we can easily detect it by its effects.

The value of a force, whatever it be, may be represented by the number of units of weight, for example the pound and its derivatives, by which it can be counterbalanced.

17.—Every means whereby motion is communicated to a body is called a *motive force*, or simply a *motor*. Any body moved by such a *motor* is called a *mobile*.

Quantity of Work or Mechanical Effect.

18.—The *quantity of work* obtained from a machine, that is, the *useful effect*, or otherwise the *mechanical effect*, is equal to the sum of the powers diminished by the sum of the resistances.

Power.

19.—Generally, every force causing motion is called a *power*, and every force, active or passive, opposing that motion, is called a *resistance*.

Powers, in Horology, are generally of two kinds; *weights*, which act in virtue of gravity or terrestrial attraction, and *springs*, whose action depends upon the physical laws of elasticity.

Resistance.—Different kinds in Horology.

20.—As we have already seen, every cause which opposes the motion of a body is a resistance.

The kinds of resistance which affect the body when at rest are *adhesion*, *inertia*, *pressure*, *viscosity of oil;* and when in motion, *friction*, *the air*, and, in general, every influence acting on the moving body in a direction contrary to its motion.

The detrimental effect of adhesion will be pointed out subsequently (**35**); inertia also will be considered in a separate article (**29**).

21.—The resistance of the air is proportional to the surface of air displaced. A change in its density, therefore, causes the value of this resistance to vary.

Although changes in the density of air may be neglected when considering slowly moving bodies, such is not the case with those that move rapidly. But since it is usual in Horology to give to moving parts forms which displace but a trifling quantity of air, its influence is reduced to a minimum, and, until the reverse is shown to be the case, may be treated as inappreciable in ordinary horological appliances.

22.—Friction and pressure are, in machines, the most detrimental forms of resistance. The first diminishes according

as the surfaces are more and more smooth, and, conversely, increases as they become more rough.

23.—If an oily substance be interposed between the rubbing surfaces, an amount of motive power, equivalent to the excess of resistance before its application, is set free. This statement is strictly true for Mechanics in general where both forces and pressures are more or less considerable; but the law appears, from numerous experiments, to be at fault as soon as we attempt to apply it to the very slight pressures and excessive velocities met with in Horology; and this is especially so at the last wheels in the train in watches.

In such cases, the stickiness of the oil causes irregular effects of adhesion which have a certain energy as compared with the slight force which animates the mobiles. It is even capable, especially after the lapse of a short time, of producing a resistance in excess of that between two perfectly polished rubbing surfaces. This may be demonstrated on very small new watches, freshly cleaned and jewelled with real rubies, and with hard pivots; they will (providing the motive force is slight) move more briskly for some minutes with no oil on the escapement pivots, than when the oil has been there for some months.

A magnetic needle moves more freely when the point on which it rotates is dry, than when oil is applied.

In considering the action of delicate mechanism, then, we cannot afford to ignore the resistance caused by the oil. We shall presently give an approximate estimate of its amount (**42**).

Momentum.

24.—When a body strikes or pushes against an obstacle, the latter is influenced by an amount of force depending on the weight and velocity of the moving body.

The power exerted is in fact equal to its mass multiplied by its velocity, and this product is known in Mechanics as *momentum*.

Momentum is thus the power possessed by a body in virtue of its motion, which enables it to neutralize, divert, or overcome any force opposed to it.

Difference between the Force exercised by a Body when at Rest and when in Motion.

25.—The influence, whether as a power or a resistance, exercised by a body in a *statical* condition, that is when at rest, is expressed by its weight in pounds or any other units.

The force which this same body would exert in a *dynamical* state, that is when in motion, is equal to its mass, which is proportional to the same units, multiplied by its velocity, or in other words, by the space traversed in a definite unit of time.

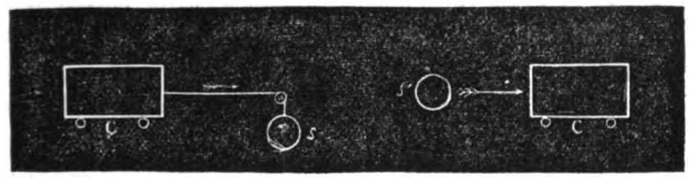

Fig. 1.

Let *s*, Fig. 1, be a metallic sphere weighing 2 pounds; it tends to move the body c with a tractive force equal to 2 pounds (neglecting the friction of the pulley).

Now let the same sphere be set in motion by an impulse lasting ten seconds, which, during the first second, causes it to traverse a space of one foot. The force exercised by the body, or, in other words, the momentum, may at the end of the first second be approximately represented by its weight; but assuming that during the last second, the space traversed be 12 feet, the sphere *s'* will strike against c with a force represented by its weight multiplied by its velocity, or 24 pounds.

What is Gained in Power is Lost in Velocity, and Conversely.

26.—Since the energy of a body in motion is equal to its mass multiplied by its velocity, the mechanical effect will obviously remain the same, if we diminish one element in the expression, at the same time increasing the other in a corresponding proportion.

If the sphere (*s'*) Fig. 1, instead of weighing 2 pounds weighs 4 pounds and strikes the body c at the instant at which it attains the velocity of 6 feet per second, the energy displayed is equal to the weight 4 multiplied by the velocity 6, or 24 pounds, identical with the preceding result.

The weight has been doubled, but the velocity has been reduced one-half, for at the moment of contact it moved during the same interval of time with only half the velocity of the former case.

27.—In considering a mechanical operation, the time occupied in its completion may be neglected: the above principles remain equally true, but the term *velocity* must be taken

to mean simply *space passed through.* It will be convenient in this case to express the above law in the following terms:

What is gained in power is lost in space passed through; and conversely, *what is gained in space is lost in power.*

Take the case of one hundred shots to be transported two miles. If each weighs 10 pounds, there will be a total weight of 1,000 pounds.

If the removal is acomplished all at once by a conveyance, the mechanical work will be expressed by the weight, 1,000 pounds, multiplied by 2 miles, the distance traversed, or by the figure 2,000.

If the work be done by a single man, carrying 2 shots at a time, the mechanical effect will be measured by the weight of the two shots, 20 pounds, multiplied by the space traversed in the 50 journeys, or 100 miles, that is, by the same number 2,000.

The useful effect or total resulting work is equal in the two cases; but in the first an extra expenditure of force diminished the space to be traversed, while in the second, where the force employed was fifty times less, it became necessary to devote fifty times more "velocity" to the operation; in other words to traverse a space fifty times as great.

Time Necessary in order that a Body set in Motion may attain a Maximum Velocity.

28.—As the energy developed by a body in motion is a product of the mass into the velocity, it is evident that the greatest effect is obtained when this velocity is at a maximum.

A body caused to move by an instantaneous force, such for instance as an ignited gas, at once attains its maximum velocity.

Such, however, is not the case if it is set in motion and retained in motion by a constantly acting influence, as for example, the force transmitted by a train of wheels to an escape-wheel. This wheel never attains instantaneously its greatest rapidity, the maximum only being reached after an interval more or less prolonged.

But the balance of a watch is, at the instant when the escape-wheel begins to influence it, moving with a previously acquired velocity; in order, then, that an impulse may be communicated to it, it is of the first importance that the contact between the wheel and the pallet of the balance

should continue until the velocity of the former exceeds, by a certain definite amount, that already possessed by the balance; that is to say, until the initial velocity of the balance has been increased in a ratio corresponding to the motive force employed and the extent of oscillation required.

Figures will render this truth more evident.

Assume 2 dwts. to be the weight which would neutralise the tangential force exercised by an escape-wheel at its circumference, after an angular displacement of five degrees of the lift has taken place in $\frac{1}{100}$th of a second. It will not then have attained its maximum velocity, and the force acting on the wheel may be expressed by the product of the weight 2 multiplied by a velocity 1, that is 2.

As the lifting action continues, the velocity of the escape-wheel will increase; and, assuming that it occupies half the time in traversing the next five degrees, the power generated is now equal to the weight 2 multiplied by the velocity 2, or to 4; that is, the force exercised by the wheel, acting as a lever, is doubled.

The calculation of the different effects constituting a lift is very complicated. The above figures must, of course, not be taken to be accurate, as they are only given in order to illustrate a mechanical law.

This subject has been specially dwelt upon because the neglect or ignorance of the fact that time is necessary in order that a body, set in motion by a continuous action, may attain a certain maximum velocity, has caused both authors and practical men to give expression to very erroneous ideas, especially with regard to the lift of the escapements and certain characteristics of such timepieces as only require winding up at long intervals.

Inertia.

Definition.

29.—The meaning of scientific terms is often in part lost when they are employed by practical men. Thus the word *inertia* is, with them, synonymous with equilibrium; a balance of a watch, a wheel, or a pair of pallets, is in a state of *inertia*, according to the erroneous language of the workshop, when that balance, etc., is equilibrated on its horizontal axis in all the positions we can cause it to assume.

Such an employment of the term is unfortunate.

Inertia is that inherent property of matter, that tendency

in virtue of which every body continues in the state in which it already exists, at rest if it is at rest, and in motion if it is in motion. It is exemplified in the excessive resistance offered by a body to being suddenly set in motion, or brought suddenly to rest when in motion.

A horse, harnessed to a heavy waggon, strains violently and makes great efforts in order to set it in motion, but draws it along with ease when this is once accomplished. On the contrary, when the waggon has attained a considerable velocity, the horse cannot stop suddenly without receiving a violent push forwards.

These two effects are due to the inertia of the mass of the waggon.

Function of Inertia in the Action of Escapements.—Heavy Wheels.

30.—Every wheel, however light it be, must have some appreciable weight; it is, therefore, subject to the law of inertia. Hence it results that when we wish to set it in motion round its axis, it cannot commence moving instantaneously; there is a period of transition from rest to motion, which, although not always perceptible, is none the less real, and the wheel only attains its maximum velocity after a certain arc has been traversed by any point on its circumference.

As the effects of inertia thus increase with the weight of the body and its velocity, it is very important to note their influence on escapements, especially during the lifting action; the wheel then travels during a very short space of time with a considerable velocity.

The following example of the influence of inertia has actually occurred in practice:

In a detent escapement with an escape-wheel fully heavy, the motion of the balance was sluggish, and the oscillation was of but moderate extent. The workman engaged on it cut away part of the interior of the wheel and reduced its arms; in short, materially diminished its weight, and, by this simple change, very appreciably increased the extent of oscillation of the balance.

It is hardly necessary to explain that the heavy wheel, offering an excessive resistance to motion, supplemented the resistance caused by friction and oil; as the wheel was longer in commencing its motion, and turned more sluggishly, it did not come in contact with the impulse pallet until after the latter had traversed a considerable portion of its angular path. The final result was a noise, and but slight impulse.

The wheel, after being reduced, commenced its motion sooner, and, almost immediately coming in contact with the pallet, accelerated its motion to the requisite extent.

Errors with regard to Light Wheels.

31.—From observations analogous to that above described, it is generally assumed, and set down as a mechanical truth, that in every escapement the wheel should be as light as possible. A question which has not received sufficient attention has thus been decided in a very absolute manner, and the solution of a particular problem has been made binding on all the escapements used in Horology.

Would a wheel entirely wanting in inertia be a valuable acquisition? There seems great reason for supposing that it would not. But, although such a case could not occur, since the metals employed always have an appreciable weight, it is none the less useful to point out that the velocity of rotation to be communicated to a wheel depends on the manner in which it influences the pallet of the balance, and on the amount of energy it is required to give out while actually impelling the balance. The following observation of a clever watchmaker, M. Monvel, will do more to explain the subject than a considerable amount of argument, and will also illustrate the converse of the case above cited:

A chronometer escapement worked well although the wheel was somewhat heavy, but when this was rendered much lighter, it caused the escapement to catch. The excessive lightness of the wheel was evidently the cause of this fault, as it changed position more rapidly than the balance; that is to say, instead of coming in contact with the face of the pallet when it had had time to come to a suitable position, the wheel commenced moving with considerable rapidity and struck the angular extremity of this pallet, producing a butting action. Every watchmaker is aware that a slight displacement of the impulse pallet is all that is required in order to avoid such a stoppage, and that the above case is only quoted as an example of the influence of inertia.

32.—Experiment, and a consideration of the nature of the metals employed in practice, show without doubt that, in those watches in which the vibrations are rapid, it is necessary to make the escape-wheel *as light as possible*, but care must be taken not to unduly diminish its solidity.

The word *solidity* does not here merely imply that the wheel must resist certain causes of breakage or distortion; an escape-wheel must be absolutely *firm* throughout, and this firmness can only be secured by care in the choice of the metal employed, and of the form given to the wheel. Thus, an arm of a wheel of rectangular section is less rigid when placed edgeways than when its broader face is parallel to the plane of the wheel.

With regard to such horological appliances as are regulated by a pendulum or a heavy annular balance, it remains for experiment to ascertain whether a certain slight amount of resistance due to inertia in the wheel is not necessary, since this wheel must move with a velocity determined (1) by the greater or less inertia of a train of wheels of a definite weight which abandons its state of rest or recoil; and (2) by the velocity acquired by the arm on which the wheel acts, an arm whose motion is slow in comparison with the velocities met with in watch movements.

33.—Inertia is proportional to the masses of bodies when their velocities are equal, and to the squares of their velocities when their masses are equal.

Separation of Working Parts when in Contact.—Adhesion Effects.

34.—Working parts in contact with each other should separate by sliding action, and not by a sudden drawing asunder in a direction perpendicular to their touching surfaces, as such an action would involve the inconvenience of variable resistances, depending on the greater or less adhesion or cohesion of these surfaces.

35.—The phenomenon of dry adhesion is well known to anyone who has brought two perfectly dry surfaces in contact, which had been surfaced against each other. The explanation of this physical fact will be given in the article on *Capillarity* (**90**).

Rolling friction (which is in reality of the nature of pressure), such for example as that of a pivot rotating between two friction wheels, or of a lantern pinion when the pins are movable, or of a pair of helicoidal toothed wheels, etc., ultimately resolves itself into an adhesive action, especially noticeable in the case of small mechanism. The presence of oil increases the intensity of this effect.

At the pallets of an escapement this inconvenience can only be avoided by developing two surfaces, of somewhat unequal extent from each other, thus producing a slight sliding friction in place of the friction of rolling.

Blows often repeated on a point ultimately occasion variableness in the adhesion effects, especially when the object struck is rigid. This remark is directly applicable to the case of the bankings in the lever and detent escapements; or of balance-springs where one coil oscillates between two curb pins, etc.

When a moderate sliding friction does not suffice to neutralise the effect, it may be further decreased by making the stop slightly elastic.

The amount of adhesion between clean surfaces is difficult to determine, and it is therefore impossible to give exact information with regard to it.

36.—In the case of oiled surfaces, the resistance due to adhesion is *proportional to the extent of the surfaces in contact.*

Laws of Pressure and Friction.

37.—A body presses on the surface by which it is supported with a force equal to its weight. Hence if the weight be doubled, the pressure is double, and *vice versâ.*

Pressure is therefore proportional to the force producing it; in other words, *to the weight which would balance it.*

38.—*Friction between surfaces is in proportion to the pressure.* Thus a body sliding along a surface will encounter twice as much resistance to its motion when its mass (or weight) is doubled, and the converse is also true.

The energy of friction varies then directly with the pressure.

A badly shaped edge in the cylinder of an escapement, or a particular form given to the inclined planes of the teeth of the escape-wheel, causes the *lift* to take place under variable pressures, and, as the friction is greatest at those points which are subjected to the greatest pressures, it is at them that the wear is most considerable.

Unequal pressures occur in the case of epicycloidal-formed teeth, and thus the surfaces of contact wear each other away unevenly; the initial conditions of the depth are therefore in time entirely changed.

39.—*The intensity of friction is independent of velocity*, provided an oily substance be interposed. This law has been proved to be true in the case of heavy machinery, and Moinet has experimentally shown its applicability to horological science.

Until the reverse is demonstrated to be true, we are therefore justified in admitting that friction can be neither increased nor diminished by varying the velocity of a moving body.

40.—The pressure remaining the same, *the resistance of friction is independent of the extent of the rubbing surfaces.* If, for example, the extent be tripled, it follows that three times the number of molecular elements rub against each other; but as each individual friction is only a third part of what it was previously, the general effect is the same.

The above is simply the statement of an experimental law; its consideration will be reserved for the following articles.

The extent of surface in contact should vary with the nature of the material.

41.—The last law shows that there is no advantage to be gained by diminishing the extent of the surfaces of contact in depths, in impulse, and even locking faces of escapements, as is too often done under the impression that friction is thereby reduced. Since the same blow or pressure is withstood by a smaller number of elements it will act with a greater force on them, and will distort the surfaces more rapidly; the accuracy of their forms will thus be destroyed in a less time.

Touching surfaces should be of such dimensions as to ensure their integrity, and it may be assumed that these dimensions depend on the nature of the body and the presence or absence of oil.

A brass wheel, when working dry against steel, should have a larger surface of contact than when opposed to diamond, which is much harder and better polished than steel. This observation is equally applicable to pivot holes, which, for equal diameters, may be shorter with real rubies than with brass.

But a distinction must be drawn between dry friction, such as that occurring at the points of contact of a wheel with a pinion, and the friction between surfaces coated with a greasy substance. It has been seen that surfaces of the latter class are separated with greater difficulty according as their extent increases.

Laws of Friction as applied to the Moving Parts of Horological Appliances.

42.—It has been mentioned in the article on *resistance* (**23**)

that oil, when employed in horological appliances, so modifies the character of the friction at the end of the train, where the motive force is slight and the velocity great, that the general law is not strictly applicable, although losing none of its exactness. This statement requires some explanation.

Resistance to sliding is in reality composed of two kinds of resistance; friction, properly so called, and adhesion. Friction is proportional to pressure, and independent of the extent of the surfaces; but the adhesion between greased bodies depends on the extent of the surfaces in contact.

With heavy machinery, great pressures, and, consequently, considerable forces are brought into play; and since the hardness and degree of polish of the surfaces cannot be increased in proportion to the weight, the extent to which bodies are *forced into* one another increases. A greasy substance, when interposed, behaves like an infinite number of microscopic rollers; thus the friction between clean surfaces is so enormous in comparison with the adhesion of oiled surfaces, that the adhesion effect is completely masked by the friction, and a greasy substance interposed, far from causing a loss of power, facilitates the sliding action. In such cases, daily experience proves the truth of the law that the resistance of friction is proportional to pressure.

But the case is not the same with the delicate mechanism of escapements, where the force employed is slight, and the velocity very great. The resistance to movement is doubtless, as in the former case, the combined effect of friction and adhesion; but there is an important difference due to the fact of the pressures being slight and the surfaces hard and highly polished. The friction has but an insignificant value, while the adhesion, occurring as it does between bodies perfectly *polished* and influenced by very minute forces, is supplemented by the viscosity of oil and resistances due to capillarity; these facts bring about the converse of what occurs in heavy machinery, and the *effect of adhesion* completely overpowers the *effect of friction*, properly so called. We thus conclude that, in this case, the measurement of the adhesion is the nearest attainable approximation to the resistance which opposes motion.

But since the resistance caused by adhesion is in proportion to the extent of the surfaces in contact, we can draw an *a priori* conclusion, confirmed by a large number of practical observa-

tions, but nevertheless not professing to be as exact even as a law partially confirmed by experiment, that:

43.—At the *extreme moving parts* or escapements of watches, where the use of oil is essential, the resistance of friction may be taken to be approximately *proportional to the extent of the rubbing surfaces, and to the diameters of the pivots* (**133**).

In the *intermediate moving parts* or train of wheels, which more nearly conform to the conditions of heavy machinery, this resistance *depends (in varying proportions) on the extent of the surfaces in contact and the pressure.*

And, finally, the *prime movers*, or barrel, very nearly conform to the law enunciated in paragraph **40**.

As bearing on the above it has been observed, amongst other facts, that the pivots of a fusee, although of unequal diameters, wear the pivot holes sensibly to the same extent; this result is attributable to the equality of the resistances on the two pivots.

Observations on the Laws of Friction.

44.—The data above given are sufficiently exact for horological purposes; for, on points on which it is impossible, at least for the present, to establish an absolute and unvarying law, owing to the fact that inappreciable differences in the material and method of working it influence the action of the finished mechanism, a rule giving approximately the values of resistances is a desirable acquisition. Such a guide keeps the mind constantly on the alert, and helps to make it bear fruit.

We will conclude this article by a brief enumeration of experiments recently made on the railways by certain engineers, in particular, M. Bochet. They confirm the accuracy of the laws of friction, deduced from the results of Coulomb and General Morin, only with regard to *velocities under* 4 *metres* (13 feet) *per second.* Beyond this point, and for velocities up to 25 metres (80 feet) per second, M. Bochet finds that sliding friction diminishes as the velocity increases according to a law which he gives.

Logically, it seems that friction ought to vary when a particular velocity is attained, for on becoming considerable, it evidently modifies the action between the surfaces. Thus when a soft iron disc rotates slowly, it is reduced by a file without damage to the latter; but if its velocity be very great, the file itself is attacked.

If additional experiments prove the truth of M. Bochet's theory, we shall be led to the following curious conclusion: that what has formerly been accepted as a true principle in Horology was directly opposed to the actual fact.

Percussion.—Impact.

45.—The word *Percussion* means simply the action by which a body is struck.

The centre of percussion of the striking body is the point in its mass where it acts with the greatest force.

In a bar of iron falling horizontally, the centre of percussion is in the middle of the bar; but if it be wielded in the hand, this point is at a distance of about two-thirds of the length of the bar from the hand.

In a hammer the centre of percussion corresponds very approximately with the centre of gravity of the head.

In a compound pendulum it is usually situated slightly above the middle of the bob.

In the common annular balance it may, provided the circumference is sufficiently heavy, be on the inside face of the rim. It is brought towards the axis of rotation by diminishing the weight of the rim in comparison with that of the arms.

When fitted with heavy screws, or other masses of metal projecting outwards, the centre of percussion may be found to be outside this rim.

46.—An *impact* (or drop) should in all cases be as slight as possible, as both theory and experience show this to be a source of wear and irregularity; and, as bearing on this subject, it is important to remember that physical law which asserts that every action excites a reaction of equal intensity. This reaction shows itself in the form of jerks, strains, etc., which have none the less influence on the condition and stability of the moving parts from the fact that they are invisible to the eye.

Besides causing a molecular alteration at the points struck, they occasion a loss of force, that is a diminution in the amount of useful work done. In an escapement in which the *lead* only occurs during one-fifth of the lifting angle, and a *drop* during the remaining four-fifths, less oscillation of the balance is produced than if the lift took place entirely without drop, and with a progressively increasing action normal to the pallet. In

the latter case force is rendered available, which in the former is lost in elasticity, stronger pressure, shaking of pivots in their holes, increased resistance due to the inertia of balance and its spring (which resist a sudden acceleration of movement), and finally in conversion into heat owing to the percussion or sudden vibration of the molecules.

Volume and Mass.—Density.

47.—The *volume* of a body is the amount of space which it occupies; its *mass* is the quantity of matter it contains, that is, the number of heavy molecules composing it.

A piece of india-rubber much extended possesses a certain volume and a certain mass; if forcibly compressed, its volume is considerably diminished, it occupies less space, but its mass remains the same, for it always contains the same number of heavy molecules.

The *density* of a body is the weight of a particular volume of that body. A cubic centimetre of distilled water weighs one gramme; a cubic centimetre of cast lead weighs 11·4 grammes. The density of lead is therefore about eleven times that of water.

The unit of weight employed in measuring the density of bodies is, on the metrical system of weights and measures, a cubic centimetre of distilled water at its maximum density, that is at 4° C. (39·2° F.) This weight is the gramme, the unit of weight on that system. [In England the standard or unit of weight is the avoirdupois pound, and the standard gallon of distilled water at 62° F. (16·6° C.) weighs ten of such pounds, occupying a space of 277·123 cubic inches.—Tr.]

Gravity.—Difference between Mass and Weight.

48.—*Gravity, heaviness, or terrestrial attraction* (**126**) is the tendency inherent in all bodies to fall towards the centre of the earth. The force required to prevent a body's fall is equal to its weight. The *centre of gravity* of a body is a point in its mass such that, if suspended therefrom, the body will remain in equilibrium, and the pressure it exerts on this point will be a maximum. The centre of gravity and the centre of percussion are sometimes identical and sometimes far apart from each other. In a body falling freely they are coincident; in a body rotating on an axis, such for example as the balance of a watch, the energy of percussion is in the rim, whereas the

centre of gravity is at the centre of figure. We shall be under the necessity of reverting to this subject, chiefly when we come to consider the pendulum and the annular balance.

49.—The words *mass* and *weight* are too often assumed to be synonymous; it is, therefore, of the highest importance to carefully explain their true scientific significance.

Mass, as already explained, is the quantity of matter, or rather, of molecules, which constitutes the materiality of a body. But, since heaviness is a force impelling each of these molecules equally to fall towards the centre of the earth, it is evident that *weight* is simply the resultant of all the impulses exerted by gravity on the several molecular elements. Weight is necessarily proportional to the number of molecules; we may then, when considering two bodies in the same locality, employ the words indifferently as terms of comparison.

We say purposely "in the same locality," for it is well known that the force of gravity diminishes as we recede from the centre of the earth, and a body weighed by a spring balance at the level of the sea causes a greater distension of the spring than if the weighing takes place at the summit of a lofty mountain.

This example shows at once the distinction which must be drawn between mass and weight; for we see that the mass has remained constant, as must be the case, since the number of molecules has not varied, while the weight has become less. This variation, moreover, must occur, since the force of attraction diminishes as we ascend above the sea level.

50.—The mass of a body will be known as soon as the intensity of the force of gravity has been ascertained; and since all bodies fall in a vacuum with the same velocity, the determination of this velocity will give us a measure of the force of gravity.

If the weight of a body be known in kilogrammes, its mass may be found by dividing this weight by 9·81, the velocity in metres which a body acquires after falling in a vacuum during one second of time. [On the English system the mass would be found by dividing the weight in pounds by 32·2, the equivalent in feet of the above number of metres.—Tr.]

51.—Errors will be avoided by observing that, in order to facilitate calculations, we may indifferently employ either masses or weights according to circumstances, since they are

always proportional. The mass or the weight of a body may also occasionally be taken as a measure of its power; a measure which, in such a case, is only relative and useful for purposes of comparison. It only gives an absolute measure of its value when this power could, at the instant under consideration, be exactly balanced by this mass or weight.

This is the sense in which the figures employed in articles **24** to **29** are to be regarded. These figures are not as mathematically accurate as the matter in question demands; for in practice some of the propositions, given there in very simple forms, become complex, and for their accurate solution involve abstruse problems connected with *vis viva*, etc. To fully consider them, we should be compelled to introduce arguments too much advanced for the majority of young watchmakers, and therefore useless. The object in view before all else is to instil into their minds a notion of the nature of mechanical forces.

Centrifugal Force.

52.—*Centrifugal force* is that cause which tends to withdraw a rotating body from the centre round which it revolves, and to disconnect the several parts of a body itself. Should the cords, or whatever retains it in its path break or be deficient in power, the body flies off in the direction of a tangent to the circle which it was describing (**72**). This is the case with a stone projected from a sling; with a drop of oil placed on the tooth of an escape-wheel, when the train is allowed to run down, etc.

The study of the effects of centrifugal force will form an important part of the portion of the work which treats of balances.

The Lever.

53.—As usually understood, a lever is an inflexible rod serving to lift or carry any given mass, or to change the direction in which it is moved.

Let R (fig. 2) be the mass which it is required to lift. It is well known from experience that if the extremity of a bar R P, be introduced below R, and a solid fulcrum be placed at A, the mass R will be lifted with ease, whatever its weight may be, either by suspending a weight at P, or by a pressure of the hand at that point, provided that the length of the arm A P is very considerable in comparison with that of A R.

54.—This illustration shows that in every lever there are three points to be considered: (1) the point of support or fulcrum A; (2) the point R at which the resistance acts or where the weight to be raised is placed; and (3) the point of application of the power P, employed to overcome this resistance.

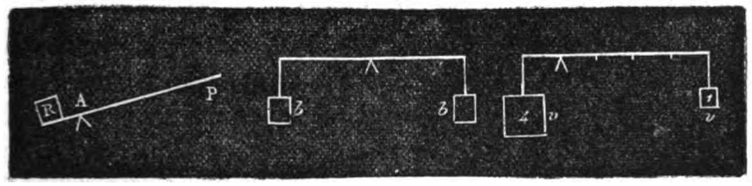

Fig. 2.

The lever may thus be divided into two parts, A P and A R, termed *arms of the lever*, which are distinguished by the purposes to which they are applied, one being called the power arm or lever (A P), and the other the resistance arm or lever (A R).

55.—When the two arms are equal, it is necessary to suspend equal weights *b b*, from their ends, in order that they may neutralise each other, for the power is then equal to the resistance.

The best known example of this case is the common balance, where equal weights counteract each other. But if the arms are of unequal length, for example the one four times the other, a weight of one pound placed at the extremity of the longer arm will be sufficient to balance a weight of four pounds suspended from the extremity of the shorter. The power and the resistance are now equivalent, for each is ascertained by multiplying the weight by the length of its lever arm.

An example of such a form is met with in the Roman steelyard, and everyone has occasionally seen very bulky objects weighed by this means. A sack weighing several hundredweight, for example, is suspended from the short arm of the lever and can be counterpoised by a one pound weight hung from a point on the longer arm which is ascertained by trial. The number of pounds in the weight of the sack is at once ascertained when we know the number of times the length of the shorter arm is contained in that of the longer, measuring from the points of application of the two forces.

56.—These examples will be sufficient to demonstrate this principle of Mechanics; *in every lever, wherever the fulcrum be*

situated, the power is to the resistance in the inverse ratio of the lengths of their arms, or, in other words, if one arm is one-half, one-quarter, etc., of the other, it will be necessary, in order to secure equilibrium, that the weight suspended from the extremity of the longer arm be one-half, one-fourth, etc., of that suspended from the shorter arm.

57.—It should be observed that, in calculations with regard to levers, we must take the unequal weights of the arms into account, as well as notice that the position occupied by each of the three points, of *support*, of *resistance*, and of *power*, may vary; indeed, this variability of their positions is the basis on which levers are subdivided into *three classes*. But the first of these observations is not applicable to the case of escape-wheels, as they are assumed to be balanced on their axes; and since the second does not in any way modify the principle above laid down, we can conveniently defer the further consideration of the subject of *Levers* to the *Introduction* to the study of depths.

Application.

58.—An escape-wheel is a lever in equilibrium on its fulcrum. The *leaf* of the pinion rivetted to this wheel is the arm to which pressure is transmitted from the source of power, and the *radius* of the wheel is an arm transmitting that impulse to the escapement, a lever of impulse generally known as the *pallets*, being interposed.

In relation to the balance, the radius of the wheel is a power arm and the pallet on which the wheel acts is a resistance arm.

The power exerted by the wheel is opposed by the resistance of the balance with its attached spring, and the two forces are in equilibrium when the power is to the resistance as the radius of the wheel *b a* (fig. 3) is to the length of the impulse pallet *a c*.

If the arm *b a* were reduced in length by one-half, equilibrium could only be maintained by doubling the amount of resistance opposed to the wheel; and conversely if the reverse were the case.

59.—Relying on this principle, which, while in itself unvarying, receives a false interpretation at their hands, some watchmakers are continually assuming that reducing the

diameter of the wheel by, say, a half, will suffice to produce an increased vibration of the balance, since the force applied to it would thus be doubled. They are surprised when a trial shows the fallacy of their mistaken theory, a theory which forgets, among other points, that when a body passes from the statical condition or state of rest to the dynamical condition or state of motion, a new element, velocity, must be considered; and that this velocity, when existing during a very short interval, such as that required for a single lift of the escapement, is sufficient to introduce sensible differences, especially in regard to the promptitude with which the motion commences.

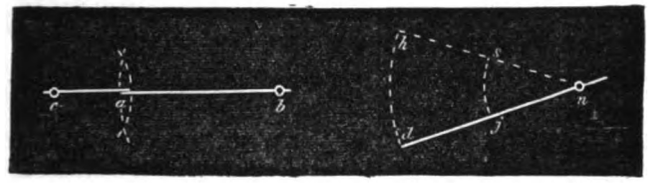

Fig. 3.

60.—Given any watch escapement, the mechanical effect produced by its wheel, remains *theoretically* the same, however we may vary the diameter of that wheel.

Let *d n* (fig. 3) be the radius of the wheel, *d h* the arc described by *d* while an impulse is being communicated to the balance. Halve the diameter, and the radius becomes *j n*, and the arc described *j s*, is only one-half of *d h*. In the former case the force was one, and the space through which it acted two; in the latter, the force was two and the space one. Mechanical effect is expressed by the product of force into velocity, and therefore the result is, theoretically, the same in the two cases we have been considering (see **120**, **124** and the end of **51**).

But what is the exact meaning here of the expression, mechanical effect? Simply this: that the *work* done by the wheel was the same in the two cases; but it by no means follows from this that the *resultant* of this work was entirely available in both cases for producing the oscillation, or that its extent was the same in both.

61.—We have said "theoretically" above, for in no case can the entire force of the wheel be communicated to the balance, as this would require all the working parts to be independent of the laws of friction and inertia, which is impossible. Almost

every change in arrangement, size, form, etc., involves a variation in the intensity of these two resistances.

62.—We must, then, first calculate the amount of the mechanical effect, and then apply a correction to it, on account of the resistance occasioned by friction and inertia, before coming to any conclusion with regard to it; for, otherwise, all inferences of a theoretical nature will be falsified by practice.

The Inclined Plane.

63.—The force required to cause a body to slide when placed on a horizontal surface, must be equal to the resistance opposed by friction, since the action of gravity is neutralised by the support.

The force requisite in order to cause a body to ascend an inclined plane increases with the inclination. In this case it is necessary to overcome both friction and that part of the action of gravity which draws the body towards the base of the plane.

If the plane becomes vertical, the friction is reduced to zero; but it is necessary that the power neutralise the entire force of gravity, that is to say, the power must be equal to the weight of the body before it suffices even to balance it.

The resistance offered by a body sliding or rolling down an inclined plane is, therefore, due to two principal causes: gravity and friction.

A Body Raised or Supported on an Inclined Plane.

64.—When the power is expended in raising the mass R up an inclined plane *a b* (fig. 4), and acts parallel to the plane: *the power is to the resistance as the height (b c) of the plane is to its length (a b).*

Assume the body R, weighing 1 pound, to be drawn in the direction *a b* while resting on an inclined plane 8 feet long, the height *b c* of which is 2 feet; it will require a weight P, equal to one-fourth of a pound, to counteract the resistance of the weight R, and this will then remain at rest on the inclined plane.

From this it immediately follows that, if the length *b a* remain constant, but the height *b c* be reduced one-half, the weight R can be maintained at rest by a force equal to one-eighth of a pound, and *vice versâ.*

65.—The power may act horizontally, that is, in a direction parallel to the base of the plane; in this case, *the power is to the resistance as the height (g d) of the inclined plane is to its base (g f).*

As an application of this we will quote a case of frequent occurrence in Horology.

Take a movable inclined plane, that is, one which can be caused to travel backwards or forwards.

Suppose that a force P tends to move it from *g* towards *f* (fig. 4), while an obstacle or resistance R, opposes its motion.

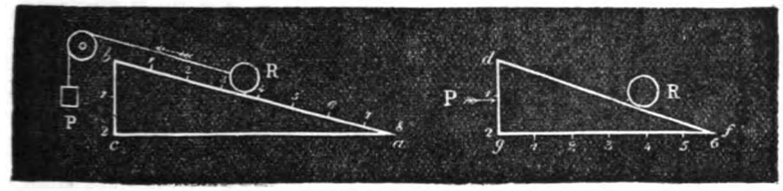

Fig. 4.

Let *g f* be 6 feet in length, and *g d* be 2 feet high; then if the power P be taken equal to 3 pounds weight, it will require a resistance of 9 pounds, in order that the plane may remain stationary, since P and R would then be in equilibrium.

The Inclined Plane in Motion.

66.—We have thus far considered the inclined plane, subjected to the combined action of a power and a resistance which neutralise each other, from a statical point of view.

The equilibrium thus established between P and R is liable to be disturbed, in the case of an inclined plane capable of motion and of an obstacle also movable, by an addition of force, either to P, when it will advance, causing R to ascend, or to this resistance R itself, and then the plane will go backwards in consequence of the pressure of the body R.

There is thus a change of condition from the statical to the dynamical. Let us now consider the action of the inclined plane when in motion.

If the original height of the plane be *reduced* by one-half, the power when reduced in the same proportion will still suffice to move the opposing body through a space equal to half that which it traversed when the first plane was employed.

Conversely, if the inclination of the first plane be *doubled*, the base remaining the same, it will require twice the force to drive the body through twice the distance.

The gain in power, therefore, of the first case, is accompanied by a reduction in the distance travelled; and in the second, the gain in distance travelled involves an increased expenditure of force.

The mechanical effect of the two inclined planes is the same. It appears, then, to be a matter of indifference which is employed in an escapement. This conclusion is quite true theoretically—that is, if we ignore the opposing forces of friction and inertia; but to arrive at a practical conclusion we must correct it for these two resistances. It is identical, in fact, with the case considered in paragraphs **60**, **61**, and **62**.

Composition and Resolution of Forces.

67.—In a train, the motive power is transmitted in its entirety from the first to the last moving part.

The force exerted by an escape-wheel at its circumference, represented by the weight which it can neutralise, is also transmitted undiminished through the several parts of the escapement in connection with it.

Nothing is lost in nature, but all is continually being decomposed and recomposed; and force—that influence whose effects we see without knowing its cause—is, like the entire universe, subject to this law.

The action of a prime mover, as it is transmitted step by step through a train of wheels, divides itself into two parts: one overcomes the resistances opposing its action, and the other is that part of the force which is free, and therefore available for producing any useful or desired effect.

The less this decomposition or resolution of the force, the greater is the useful effect which it can produce.

68.—The resistances to be overcome by an escape-wheel during the act of lifting are due to two principal causes: friction, which, as already shown, varies with pressure, and inertia, proportional to mass.

To make this clear by an example, let the circumferential power of an escape-wheel be equivalent to 5 dwts.; and let the friction and the resistance caused by inertia in the balance require a weight of 3 dwts. to neutralise them; the power of the wheel is thus resolved, and the useful effect, as shown by the extent of oscillation of the balance, is equivalent to a weight of 2 dwts.

Suppose, now, that, without changing at all the train of the watch, by a change in the arrangement of the balance, or by better forms or better workmanship, the resistance due to friction and inertia during the lift are reduced to 2 dwts., the

effective action of the wheel on the balance will be increased by one-half, and the amplitude of the oscillation will increase in consequence.

Parallelogram of Forces.

69.—The power available when two forces act in directions opposite to each other is equal to their difference.

When a body is set in motion by two forces converging to a point, the following geometrical construction gives the intensity and direction of the force impelling it.

The method is generally referred to as the graphic solution of the problem of the *Composition of Forces*. A complete explanation of the principles on which it is based would involve considerations too advanced for an elementary work; its simple enunciation, therefore, followed by an illustrative example, must suffice.

Consider the two forces as being applied during the same interval of time, one second.

Let one force of 10 pounds act on the body a, in the direction $b\,a$ (fig. 5); and let the other force of 6 pounds, act on a in the direction $c\,a$. Divide $a\,b$ into 10 equal parts, which we will take for our units of length, and set off on $a\,c$, commencing at a, six of these units. From the point b draw $b\ d$ parallel to $a\ c$, and through c draw $c\ d$ parallel to $a\ b$; we have thus constructed the parallelogram $a\,b\,d\,c$. The line $a\,b$ represents the direction of a force of 10 pounds, and $a\,c$ that of a force of 6 pounds. The diagonal $d\ a$ of the parallelogram represents the direction in which the body is impelled, and its length, measured in the units above referred to, shows: firstly, the number of pounds pressure to which a is subjected; and, secondly, the space that will have been traversed by the body in the time it would have required to traverse the path $a\ b$ or $a\ c$, under the influence of one or the other force alone.

If instead of forces expressed in pounds, the velocities which they communicate to a had been given, the solution would have been identical.

Let us now assume that the body a is moving in the direction $a\ y$, in consequence of an initial impulse communicated to it, and that it meets with opposition to its motion acting in the two directions $a\ x$ and $a\ z$; the parallelogram $a\,x\,y\,z$ will enable us to ascertain what portion of the initial impulse is employed

in overcoming each of these resistances, providing that the line $a\,y$ indicates both the direction and the intensity of the power acting on a.

Application.

70.—A body, set in motion by the extremity of a lever rotating on an axis, receives from this lever an impulse or pressure in a direction perpendicular to it.

Let $f\,g$ (fig. 5) be a lever, or the radius of an escape-wheel in contact with a cylinder capable of rotating on its centre; represent the force exerted by this lever, and the perpendicular direction in which it acts by $g\ h$. In this case the cylinder pivots are subjected to a direct pressure from the tooth of the wheel, and the pivots of this wheel are not strained.

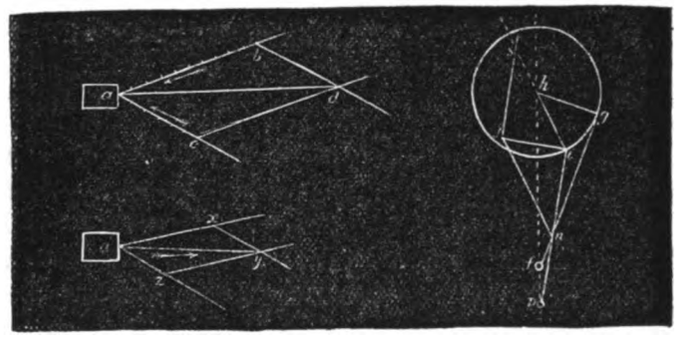

Fig. 5.

Now transfer the point of contact to i. The friction on the cylinder will be increased, and the pivots of the escape-wheel will be subjected to pressure. What, under such circumstances, will be the amount of the friction?

Neither the force exerted by the lever, nor the direction in which it acts, has been altered; and they may be represented by the line $i\,j$ equal to $g\ h$. Through j draw $j\ n$ parallel to $h\,i$, and through the same point draw $j\ s$ parallel to $i\ n$, and we shall obtain the parallelogram $n\,i\,s\,j$, whose side $i\ s$, measured on the scale on which $i\,j$ and $g\ h$ were drawn, gives a measure of the increase of friction on the cylinder. The side $i\ n$ shows the amount of the pressure exerted on the wheel pivots.

A simple comparison of the two lines, $g\,h$ and $i\,s$, is sufficient to show that the friction on the cylinder increases rapidly as the point of contact approximates to the line $h\ p$, and that, as

the power is more and more resolved, a less amount remains available for performing useful work.

71.—An intelligent reader will perceive without further explanation, that a graphical construction, analogous to that given above, will indicate at once the amount of the force exerted by a lever which acts against another lever; such, for example, as the action of an escape-wheel tooth against the impulse pallet. But in this case the point at which the force is normal, that is, perpendicular to the arm receiving the impulse, is on the line which passes through the centres of movement of the two levers, corresponding to the line *h p*.

The following article will make this question clearer; it may, by some readers, be considered to be somewhat of a repetition, but we are very anxious to explain the question to practical men.

On Tangential Escapements.

72.—A straight line *a b* (fig. 6) drawn perpendicular to the extremity of the radius *a c* of a circle, is called in Geometry a *tangent* to that circle; it can evidently only touch it in the one point *a*, however much it be extended in either direction.

It appears necessary in the first place to give this definition because some authors, having a practical knowledge of Horology, but ignorant of Geometry, often confuse the tangent with the *secant*, another line, which, on being produced, cuts the circle, and, therefore, meets it in two points (*p d*).

Tangential Impulse.

73.—Theory indicates that an impulse can be most advantageously given to a body rotating on an axis by applying the force in a direction perpendicular to the arm of a lever which passes through the centre of rotation; that is, along a tangent to the circle of which this lever is a radius.

The impelling force under this condition is communicated, in its entirety, to the body, and the impulse which it receives is, therefore, the greatest that can be communicated to it by the given motive force.

Readers deficient in the necessary theoretical knowledge on the subject can easily demonstrate the truth of this fact by turning the handle of a wheel, a grindstone for example; they will soon find that they command the greatest amount of power, either in pushing or pulling, with the least effort, when

the force is applied at right angles to the arm $b'c'$, and this arm

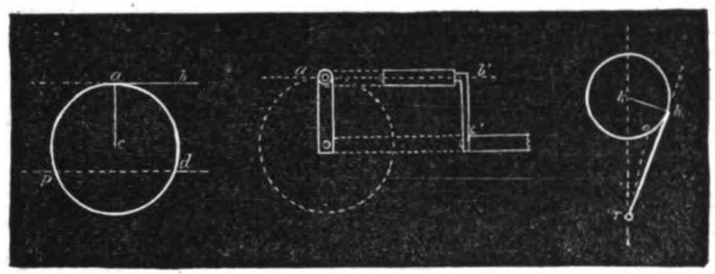

Fig. 6.

is simply a radius of the circle described by the handle. Hence, at these points the power is applied in the direction of a tangent to the circle.

Tangential Rest.

74.—When the arm of any lever, such as hr (fig. 6) presses against a circular body, which has an alternating motion of rotation round a centre k, the position involving the least amount of friction is, as we have already seen, that shown in the figure, namely, when the line drawn from the centre of movement of the lever to the point of contact is a tangent to the circle. For, whether the cylindrical body turn to the right or left, the lever acts in the same manner without straining its axis, and the nature of the friction is similar in the two cases, being of the kind known as disengaging friction. The resistance opposed by the lever to the motion of the cylinder is simply such as is due to its direct pressure.

If the lever be now made to rest at the point o instead of h, the state of the case is altered (**70**). This occasions an increased adhesion of the balance and escape-wheel pivots as they are strongly pressed against the sides of their pivot-holes. The friction therefore, being proportional to pressure, must be increased. An appreciable quantity of the force required to impel the balance is absorbed by this augmentation of the resistance, and the amplitude of its oscillations must be proportionately decreased.

75.—To sum up then, the further we displace from the tangential position the points of *lift* and rest, with so much the less ease will the moving parts travel, and the more rapid will be the destruction of surfaces in contact. And the inevitable results must follow; harsh friction, a falling off in

impelling force, and a necessity for employing a motive power greater than that which would have sufficed had the point of application been tangential, or nearly so.

Danger of Purely Geometrical Solutions.

76.—It has still to be shown, for each particular form of escapement, what positions the several parts should occupy in order to satisfy the above condition. This will be done in the course of the volume. We shall also show in what cases this requirement should be secondary to other and more important ones; for the recommendation of some authors that an escapement should always act tangentially must not be taken literally, but, in common prudence, must be conditional. For the escapement to be tangential, when feasible, is, without doubt, good; but this would not of necessity imply a superiority in the matter of timing. Thus, amongst many other examples, the duplex escapement may be noted, which, notwithstanding the unsatisfactory nature of its rest, can be timed to a far greater nicety than the cylinder escapement; for, in the duplex, the weight of the balance, its velocity, and the amplitude of the arcs it describes, etc., counteract the disadvantage of an untangential rest. To be exact, then, the only general statement that can be made is that *the more the points of contact in an escapement approximate to the tangential position, the less force is lost by friction, etc.*

77.—The watchmaker who would prefer one form of escapement to another, solely on account of its possessing geometrical advantages, would prove himself ignorant of the actual condition of his science, and lay the way to future difficulty and annoyance. An escapement, be it remembered, is a problem involving several *unknowns*, and to truly solve it they must all be determined. In Mechanics, as in all else, it is a very great mistake to regard a question from only one point of view.

Mechanical Complication.

78.—A complexity of action and a multiplication of moving parts is nearly always a fault, for it increases the liability to variation by rendering resistances, drops, contacts, adhesions, frictions, etc., more numerous. When we remember, moreover, that some points of contact are more liable to lose their shape than others, that the molecular state of rubbing surfaces varies according to the pressures, temperatures, etc.,

to which they are subjected, that the fluidity of oil is continually changing, and that thick bodies are not so rapidly affected by heat as thin ones, we see that in complicating machines for the measurement of time, we are but multiplying the causes of variation, and that it will be difficult, if not impossible, to estimate their relative intensities, and so find means of compensating for them. Indeed, an *elegant and efficient simplicity* will always, in escapements, as much as in general mechanism, be the true measure of genius and the acme of perfection.

THEORETICAL AND PRACTICAL CONSIDERATIONS.

CONCERNING

The moderator and regulator.—The retention of oil at contacts.—The lead.—The lift; lifting arc, supplementary arc.—Length of the impulse and locking pallets.—On the nature of the several kinds of friction, useful friction, engaging and disengaging friction.

The Moderator and Regulator.

79.—In clocks where the escapement is connected with a pendulum, loaded with a heavy bob on which terrestrial gravity acts with a uniform force in virtue of the laws of *gravitation*, this pendulum is at once a *moderator* and a *regulator;* it is evident that the mode of suspension must be such as to secure to it the greatest possible freedom and continuance of motion, and that the train should only be called upon to restore, at each oscillation, the slight amount of force that has been dissipated, and thus to interfere as little as possible with this gravitation action, the most uniform power accessible to us in nature.

80.—But the case is entirely different with the annular balance of a watch. Gravity does not in any way tend to maintain a regularity in its motion, and it thus became necessary to seek elsewhere for a regulating power; such was found in the elasticity of the *balance-spring*. Hence it follows that the annular balance must rather be regarded as the *moderator*, and the balance-spring as the *regulator*, although both, in varying degrees, share in these two qualities.

We refer to special articles, headed *Annular Balance; Balance-spring*, for details concerning the regulating power, diameter and weight of balances, as well as for theoretical and practical considerations with regard to the balance-spring. Each

of the subjects requires careful study, and the same may be said of the *Pendulum* (see under these headings in Part III.).

Retention of Oil at Contacts.

81.—It is essential to employ oil in escapements, if not on all the surfaces of contact, at any rate, on the greater number. The general arrangement and the form of the rubbing portions should be such that the oil may collect in sufficient quantity, remaining in such places as it is required while it is not drawn on to contiguous portions. A moderately careful study of the main laws of hydrostatics will show how these advantages can be secured in practice.

Movement and Equilibrium of a Liquid.—Attraction.—Capillarity.

82.—A liquid is subject to the action of three forces:—

1. Gravity, or terrestrial action.

2. Adhesion, or the mutual attraction between the liquid and the substance of the vessel containing it.

3. Cohesion, which unites the liquid molecules themselves; or, in other words, the attractive force existing amongst them, and opposing the subdivision of the mass.

If a small drop of oil be placed on a plate which is then inverted, the drop retains its position because the adhesion of the drop to the plate and the cohesive force acting between the several molecules are sufficient to overcome the earth's attraction.

If the bulk of the drop be increased while the surface to which it adheres remains the same, it will lengthen on the reversal of the plate, and then break apart; one portion adheres to the plate as a thin layer, and the other falls to the ground. The fall is occasioned by the force of cohesion being overcome by gravity.

When a drop of liquid is attached to a needle point, it at once ascends towards the thicker part; but if the point be hammered out or formed into an arrow head, the drop rests on it. The above facts explain the cause of this.

83.—When liquids, such as water, alcohol, oil, etc., are introduced into a thoroughly clean vessel, they do not exhibit a perfectly level surface: it is raised at the edge, as seen at *a* (fig. 7). If a tube be introduced, the liquid rises round the outside of it, and in the inside ascends above its mean level, as is seen at *b*.

These effects are due to one and the same cause, and that cause is the mutual attraction existing between the liquid and the substance in contact with it. It is this attractive force that raises the liquid against the solid surfaces. It is more energetic, and the elevation is therefore greater, according as the sides are brought nearer together. Thus the finer the bore of a tube is, the higher does the liquid column become.

This phenomenon is called *capillarity*, from a Latin word signifying hair-like slenderness.

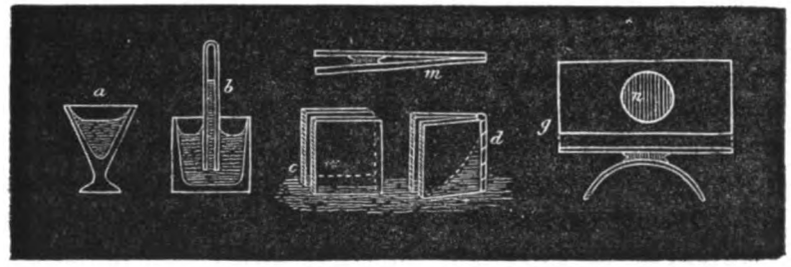

Fig. 7.

84.—Capillarity is the force which occasions the ascent of water to the upper surface of a piece of sugar moistened at its base, and of oil or alcohol to the extremity of a lamp wick; it also causes the spreading out of a drop of oil on cloth, etc. The pores of sugar and the fibres of wick or cloth act the part of so many capillary tubes.

Water ascends 30 millimetres (1·2 in.) above its external level in a perfectly clean glass tube 1 millimetre (0·04 in.) in diameter, and the height of the column increases rapidly as the internal diameter is decreased.

85.—An ascent due to capillarity also takes place between two separate bodies, say two plates, when placed close together. If their planes are mutually parallel, and perpendicular to the surface of liquid, it ascends to the same height between the plates, as shown at *c* (fig. 7). If the plates be united by a hinge, and form an angle, as shown at *d*, the height to which the liquid ascends increases as the distance between the plates decreases up to their line of junction, when it attains a maximum.

When a drop of liquid is introduced between two surfaces, arranged as at *m*, it will, if not too far distant in the first instance, gradually work its way up to the junction. Further, if a certain inclination be given to them, the liquid remains stationary, since it is then drawn equally in opposite directions, by the attraction due to the angle and by the earth's gravitation.

When the angle between the two surfaces is greater, the attraction due to capillarity is only appreciable within a very short distance of the summit of the angle.

86.—When a drop of oil adheres to two surfaces, both convex, or even one convex and the other plane, as indicated at *g*, it collects at the point, *n*, at which they are nearest together, remaining there with the greater ease according as they more nearly approach each other.

87.—Oil resting on a hard, well polished surface, spreads but little, and does not adhere firmly; but if this surface be ground, the polish is destroyed and the oil adheres better. On a metal just filed, and therefore scored by the teeth of the file, the liquid rapidly spreads, for each line is evidently a longitudinal section of a capillary tube. It should be noted that the more a given extent of surface is covered with roughnesses, the greater number of points of contact does it offer to the oil. These facts suffice to explain the reason why the chamfers made with the mandril round pivot holes, often retain a quantity of oil at the expense of the pivots, which dries, and is of no real service to the watch.

88.—Centrifugal or any other movement may overcome the force of cohesion. Heat diminishes it.

It will be evident from this why one and the same kind of oil will remain at the points of contact in one watch, whereas in another subjected to a higher temperature and to rough wear, it spreads or runs away along the axes.

89.—One essential condition must be satisfied in order that capillary attraction may occur; the liquid must *wet* the body in contact with it. When the reverse is the case, *repulsion* takes place, and the liquid instead of ascending the side of the vessel, is depressed along its entire edge, and, in a fine tube, the level is lower than that of the external liquid. This occurs in a remarkable degree between mercury and glass, for when a drop of that metal is placed on glass, it does not *wet* it, but subdivides itself into a number of small and extremely mobile globules.

90.—Capillary action is only sensible when the bodies acting on each other are in close proximity, and it not only occurs between solids and liquids, but also between two solids. The nearer the surfaces are together, the more intense is the effect.

The adhesion between clean and dry flat surfaces, referred to in paragraph **35**, finds its explanation in the above considerations.

The force with which two oiled surfaces oppose separation, when superposed, is greater according as they more exactly fit one another.

Application.

91.—The physical laws, which govern the above effects, lead to the conclusion that the retention of oil on rubbing surfaces may be guaranteed as follows:

For Pivots:—Care should be taken to leave the oil chamfers sufficiently deep, the pivots long enough and of conical shape, and the internal faces of the holes in which the shoulders of the axes rest, as well as the external faces when these holes are provided with endstones, hollowed in *tallow drop* form with a very slight interval between the bottom of the hole and the endstone. When these precautions are taken, the oil, if not present in too great a quantity, will neither spread nor run down the axis, but will remain partly in the oil-chamfer and partly attached to the shoulder of the axis, and, in the case of pivot holes with endstones, as the oil is exhausted that spread over the endstone will be drawn into the pivot hole through capillarity.

The great majority of the chamfers met with, are made without the workmen having any intelligent knowledge of the subject, and are detrimental, since they draw off and retain the oil. They should, therefore, not be made at all unless they are arranged so as to maintain the liquid in proximity with the axes, and these are then able, so to speak, to pump it up as required. An intelligent application of the laws of hydrostatics, supplemented by observation, will teach more in relation to this subject than a considerable number of illustrative examples.

For the Teeth of Escape-wheels:—Make the teeth broad and thick at the head only, in order that their mass and surface may be so considerable as to neutralise the attraction which the flat of the wheel exerts on the oil, or else make the teeth long and delicate. When such is the case, the oil is retained on the locking faces, etc.

Generally:—Surfaces of some extent, when not in contact with bodies having a much greater relative volume,

retain a layer of oil very well. Contact points that are formed with a head or spread out like a fan, deep reservoirs and chamfers (which, however, must be made with judgment for otherwise they are detrimental), and finally, the *play endshake*, and angles between the fixed and movable parts of the mechanism; if all these be so arranged as to give rise to capillary action, every precaution has been taken not only to retain the oil at the points at which it is required, but also to cause it to return thither when it has been forced away by the motion of the mobiles.

92.—This will be made clear by an example.

The geometrical principle of the hook escapement causes it to be regarded as superior to the cylinder escapement, which, nevertheless, works better and for a much longer time: for it has over the former this very great advantage, that the oil remains on the rubbing surfaces. This will be at once evident, if it be remembered that the cylinder itself presents a considerable surface where oil remains without difficulty; the teeth of its escape-wheel consist of upraised masses, separated from the flat by thin supports; and, as a consequence, the oil, unless there has been a want of skill on the part of the workman, does not pass to the flat of the wheel; and, finally, the general arrangement and the mode in which the whole acts, cause the oil to be constantly restored from the cylinder to the wheel, and conversely.

In the hook escapement, on the other hand, the oil, which must only be present in very small quantity, leaves the locking faces rapidly, and is drawn along the impulse pallet, passing partly to the back and sides of it, and partly to the teeth of the wheel. These teeth, in the form of pins fixed on the flat of the wheel, or on supports of considerable dimensions, should be very sparingly supplied with oil, and must, as a consequence, dry with great rapidity. If we endeavour to avoid this fault by putting sufficient oil to maintain its fluidity for a considerable period, it spreads itself over the face of the wheel and the same result follows; thus it happens that the escapement is, in a short time, working *dry*, or approximately so, and wear and irregularities are the inevitable consequence.

We would add a remark equally applicable to other escapements.

One cause of the rapid exhaustion of the oil in a hook

escapement is the fact that this oil is in a constant state of being driven, by centrifugal force, towards the extremity of the impulse pallet. This pallet carries it through the air, which is subjected to a considerable disturbance, and the oil is thus exposed to a true air-current, and its decomposition is thereby accelerated.

The Lead.

93.—The continuous action by which the tooth of a wheel impels the pallet of a balance or the leaf of a pinion, and causes it to traverse a definite arc of its angular movement, constitutes what is called the *lead*.

The conditions regulating its action are of the highest importance.

The lead of a balance, as well as of a pinion, is subject to the clear and well-defined laws of the theory of depths. It is by working in conformity with those laws that we ensure the preservation of the rubbing surfaces for the longest possible time, and that we reduce the interfering causes due to the resolution of force to a minimum.

We refer to Part II. *On Depths*, for numerous details concerning this question.

The Lift of Escapements.

Lifting Arc.—Supplementary Arc.

94.—The entire oscillation of a balance may be divided into two distinct portions: one during which the balance receives its impulse direct from the source of power, and called the *lifting arc*, or simply *lift*; it is by this impulse that the movement of the moderator is maintained, and its energy varies with the intensity of the motive force: the other portion, called the *supplementary arc* from the fact that the addition of it to the lifting arc gives the full extent of the vibration, is only indirectly dependent on the motive force.

95.—The lifting arc is influenced by every irregularity of this power; the supplementary arc, in great part, corrects these inequalities. The supplementary arc should, therefore, be considerable in comparison with the lifting arc.

This conclusion, although logical, must not be regarded as a principle applicable to every case. Demands are made upon us in practice which we can neither afford to neglect nor ignore, and it would be wrong to decide definitely that, of two different

forms of escapement, the best is the one which, while imparting to the balance the impulse required for the maintenance of its motion, allows of the greatest supplementary arc.

Lift is affected by inertia and friction, influences which frequently stand in the way of its period being shortened, and the counteracting effect on inequalities in the force which maintains the movement of a balance is influenced by several conditions besides the amplitude of the supplementary arc and its disproportion to the lifting arc.

Without now entering into the more advanced consideration of this subject, we would observe that the great majority of the chronometer makers of the present day, have reduced the extent of the complete oscillation from 400° to about 360°, the lifting arc remaining very nearly of the same extent as formerly.

Short and Long Lifting Arcs.

96.—The extent of the supplementary arc depends primarily on the circumstances attending the lifting action, and the difficulty of determining these is, without doubt, the cause of the divergence in the opinions daily expressed by watchmakers.

Whilst some, including many of the most skilful, take it as an unvarying principle that the lifting action should be very short and quick, that is to say, almost instantaneous, others, equally skilful, purposely increase the extent of the lift, considering that, by so doing, they perceptibly increase the oscillation of the balance.

97.—We will proceed to examine these two views, which equally run counter to the actual laws of Mechanics.

The instantaneous impulse, even if possible, would have the gravest inconveniences. It must involve violent blows, varying with every change in the motive force and the adhesion. It would occasion a waste of power, and lead to all the causes of irregularity enumerated in paragraph **46**.

As regards the increase of the lift, it is sometimes necessary when an escapement is heavily constructed. The excessive resistance occasioned by the oil, friction and inertia, renders it important that the time during which the balance receives its impulse from the wheel should be so far prolonged as that this impulse may be sufficient; but although this fact, so often corroborated by the observation of ordinary escapements,

especially those of the lever form, is true, many watchmakers have unquestionably drawn a wrong conclusion from it.

Almost every increase of the lifting angle produces further resolution of force, greater variability in the friction, etc., while it causes the lifting and supplementary arcs to become more nearly equal. But if the extent of the latter be in the first instance well proportioned to that of the lifting arc, it will, when altered, lose some of its power to correct variations in the motive force. Beyond certain limiting values, therefore, determined by intelligent observation and sound theoretical knowledge, any increase in the lifting arc could at best only pretend to compensate for one fault by another in any particular case, or in accordance with the requirements of a manufacturer.

Conditions on which the extent of lift depends.

98.—In general, the extent of arc described by the impulse pallet during the lift, or, in other words, the period of application of the motive force, necessary to produce an oscillation of given extent, depends on the friction, inertia, and the resistance caused by oil. *Reduce these obstacles to a minimum* and the required amplitude of vibration will be obtained from a lifting arc whose extent decreases as they become less.

The length of the Impulse Pallet is proportional to the diameter and weight of the Balance.

99.—Let us consider the balance of a watch, not when its motion first commences, for such an observation would be fallacious in that it would not enable us to ascertain its real effect on the escapement, but consider it in full action just after it has completed an oscillation, the watch going with its greatest regularity.

What work is required from the lifting action after the completion of this oscillation, which we may regard as the initial one?

It must simply restore to the balance the slight momentum which has been lost since the lift last took place.

100.—This being granted, let us consider the case of an axis rotating on two pivots and carrying a balance, which for simplicity may be represented by the diameter, *a d*, terminated at its extremities by equal heavy masses (fig. 8). Let the impelling

force of the escape-wheel be applied to this balance by a tooth or pallet *a b*.

Take two cases successively in which the force is applied at the points *a* and *i*, and observe what takes place.

When it acts at *a* there is no pressure applied to the pivots, and the force is entirely employed to impel the balance; the latter is visibly influenced by any variation in this power.

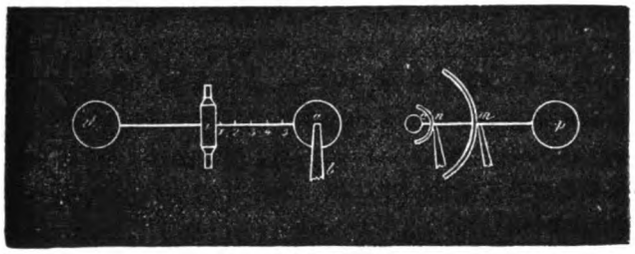

Fig. 8.

When the motive force is applied at *i*, it does not in any way contribute to the motion of the balance, but expends itself unproductively, being wasted in pressure and friction on the pivots.

If applied successively at the points 1, 2, 3, 4, 5, it is resolved; one part is absorbed by the resistance due to the pressure on the pivots, and only the excess over the force so expended is available for impelling the balance.

From 1 to *i* the pressure increases to its maximum, and the impelling force gradually vanishes. As we move towards 5, the reverse is the case, and, while the pressure diminishes, the impulse increases.

It necessarily follows from these facts that there exists a point of application of the force in the line *a i*, which would involve the least possible irregularity of movement, and it further appears that we shall practically ascertain whether we have hit the point as follows: on the one side, the pressures are too great in proportion to the regulating power of the balance and anomalies due to an excess of friction must occur. In the converse case we incur the inconvenience of a balance whose mass does not offer sufficient resistance to the variations of the motive power. (Apply the above arguments to the pendulum—**241, 1014, 1297**).

101.—Anticipating a practical application, we will compare two ordinary escapements of identical construction, except

that one balance is lighter than the other. The point of application of the impelling force should be nearer the centre of rotation in the case of the lighter balance. The same is true when two balances are of equal size and weight, but the second rotates on much thicker pivots than the first.

The diameter of the locking surface varies directly with the size and weight of the Balance.

102.—The action of the wheel in an escapement with frictional rest is always composite, or rather, it is made up of an action and a species of reaction. During the *action* the wheel accelerates the movement of the balance, and during the *reaction* (the locking) this same wheel partially checks the acceleration thus acquired.

The extent and freedom of the supplementary arc are then, to a certain extent, dependent on the intensity of what is here called reaction, and the extent of the arc varies also according to the greater or less disproportion that exists between these two successive and opposite actions of the wheel.

The reaction effects, that is, the resistances which impede the movement of the moderator during the locking, are due: (1) to the friction of the pivots against the sides of the pivot-holes; and (2) to the friction of the point of the tooth against the locking cylinder or roller.

Whatever be the radius of the roller (providing its weight remains the same), whether it be *c n* or *c m* (fig. 8), the resistance due to the friction of pivots remains practically constant so long as the pressure acts in one and the same direction.

But such is not the case with the friction at the point of the tooth, for if the force or pressure remain the same, the resistance caused by it will increase with the radius of friction.

If the locking point be too near the centre of rotation, the balance will move with such a degree of freedom that it will be affected by every change in the motive power, and will thus reproduce all its irregularities.

If the radius of friction be gradually increased (no change being introduced in the manner in which the impulse is given), the freedom, and therefore the velocity of motion, of the balance will gradually be reduced, until a point is reached at which all motion ceases.

In fact, whether the radius of friction be too long or too short, irregularity in the motion of the watch will result, for in one case the extent and in the other the duration of the oscillations is unequal. It should, however, be observed that when the radius is too short, the supplementary arc increases out of all proportion with an increase in the motive force, while the contrary is the case when this radius of rest is too long.

It follows from the above reasoning that within the space *c p* (fig 8), that is, within the length of an arm of the balance, one point and one only is so situated as to serve as a locking point for a tooth of the wheel; and further, for a given length of this arm, the point of rest is at a less or greater distance from the centre of motion, according as the rim of the balance itself is light or heavy.

Conclusions from the two last articles.

103.—As the regularity in the action of an escapement with frictional rest thus depends on a pressure whose intensity is required to vary inversely with the energy of impulsion, or nearly so, it follows that:

With a given motive force, the invariableness in the period of the movements of the balance, or, to use the trade term, the timing, depends on the securing of a definite proportion between the diameter and weight of the balance, and:

1. The diameter of its pivots;
2. The radius of the curve on which locking occurs;
3. The length and form of the pallets which receive or communicate the impulse.

The radius of the balance is the distance between the centre of rotation and its circumference of percussion. The manner of ascertaining this latter will be subsequently explained.

104.—No author, so far as we are aware, has clearly set forth this fundamental basis of the scientific construction of escapements. Had attention been sooner directed to it, observant watchmakers would doubtless have compiled tables giving the various diameters of pivots, the radii of rest and impulse, and the weight and dimensions of the balances of such watches as gave complete satisfaction. With such valuable information at hand science would have no difficulty in assigning a reason for the apparent contradictions which are observed. The question

would by this time have been decided experimentally, and even scientifically, and the watchmaker could proceed with confidence in the design and construction of escapements; a work in which he is now compelled in great measure to trust to chance.

Considerations relating to Friction.

On the different kinds of Friction.

105.—Mechanicians distinguish two kinds of friction, rolling friction and sliding friction. Watchmakers subdivide this latter kind into engaging friction and disengaging friction, but many mechanicians consider they are not justified in making the distinction. For our own part we feel that they are justified in employing separate terms since there certainly exists a difference in fact. Speaking strictly, rolling friction is not a friction but a pressure, and if disengaging sliding friction is indeed friction in the proper sense of the term, engaging sliding friction is nothing but a sort of butting.

The further consideration of this subject will be found in the introductory chapter to the study of depths, in which we shall successively study: the nature and the causes of friction; engaging and disengaging friction; the condition of surfaces during friction, and after it has occurred; and the variations in intensity of friction when accompanying a continuous or intermittent motion.

For the benefit of those who are ignorant of the subject, however, we will anticipate our remarks so far as to explain briefly what is understood by engaging and disengaging friction.

When friction occurs between two pieces moving in such a manner that one tends to force the other towards the line joining their centres of movement, and therefore called the *line of centres*, the friction is called *engaging*. Such friction is very detrimental, for it takes place, so to speak, backwards, occasioning a butting action, in consequence of which the working parts are forcibly driven apart, and thus exercise a considerable pressure on the pivots and on the surfaces in contact. Two faults are the immediate consequence; (1) rapid wear, due to the harsh friction, and (2) the necessity of applying a considerable motive power to compensate for the force uselessly absorbed by the friction.

When the driver is impelling the follower beyond the line

of centres the friction is termed *disengaging*. It is no more than a very gentle sliding action that takes place, and the pressure on the axis is much less than in the former case.

Whence it is evident that, were it possible to substitute disengaging for engaging friction during both the impulse and locking of an escapement, the destruction of the surfaces would be far less rapid, and the required mechanical effect would be obtained by using a prime mover of much less energy.

Various opinions on Friction.

106.—In every machine the resistances occasioned by friction absorb a considerable portion of the motive power. Thus, in a train of wheels, though it be well constructed, we must always consider that about a third of the force applied will be required to overcome these resistances, and, therefore, is entirely lost from the point of view of the work obtainable.

The composition of a machine is thus objectionable; from the very nature of friction it follows that the surfaces in contact mutually destroy one another; the force is transmitted irregularly through the train of wheels, and the amount absorbed in utter loss increases rapidly, occasionally to such an extent as to entirely stop the action of the motor.

The question with regard to friction, then, comes before the watchmaker under two aspects.

On the one hand: *absorption of power*, an absorption which increases as the resistance caused by friction increases.

On the other hand: *irregularity in the movement of the mechanism* whenever the nature and intensity of the friction vary.

107.—Looking at the question solely from this latter point of view, watchmakers have almost without exception concluded that, with the majority of escapements, it is preferable to render the friction (of course retained within moderate limits) smooth and uniform rather than attempt to reduce it, thereby securing questionable advantages. They are still opposed by certain of their confrères who consider friction, of whatever kind, to be a constant cause of variability and destruction. According to these latter, perfection would be attained by entirely suppressing friction in all its forms.

The first class say, "Experience shows that in numerous cases a certain amount of friction tends to neutralise the effects

of a variable motive power (**103**), which is proved by the fact that, of the escapements in use, the best regulated are not always those in which the friction is least;" and they with reason add, "that it is not so much friction itself as *its inconstancy* that is a cause of anxiety."

To this the second class reply, "Theory shows that variability in timing increases with the amount of friction, and, if the above assumption be true, it is difficult to explain the extraordinary regularity attained to in marine chronometers; for in them the friction of the escapement is reduced to the lowest possible amount, and, indeed, practically done away with during the supplementary arcs."

To the latter class we would say that it is an error to take a relative truth as an absolute truth; and, although they may be right as regards one particular case, they are wrong as regards a large number of other cases; that theory by no means shows variability in timing to increase with the amount of friction; and, finally, that in maintaining so biased an opinion they ignore the fact which skilful chronometer-makers can easily prove, namely, that escapements involving a minimum of friction can only be timed when provided with an isochronal spring and perfect compensation—expensive accessories that are useless in the case of escapements with frictional rest, but these will nevertheless, without such additions, afford a regularity which is perfectly satisfactory for all ordinary purposes.

A singular feature in the discussions which have taken place on the subject we are now considering is that those who advocate the total suppression of friction at the same time admit correcting pressures, such as are found in Graham's escapement amongst others; just as if the pressure did not, during the motion of the escapement, occasion friction.

108.—Bearing in mind the results arrived at in the articles summarised in paragraph **103**, we shall presently state the conclusions of a discussion now finally settled; and we can, before doing so, give numerous practical proofs that, in some cases, a certain amount of pressure is a necessity (**38**); we, however, confine ourselves to the following as sufficiently decisive.

Experimental evidences of the advantage of a certain amount of Friction.—Résumé.

109.—If a verge escapement be provided with jewel holes and pallets, its contrate wheel pivots supplied with endstones, or

the pivots of the balance-staff be extremely fine, all precautions tending to diminish friction, it will be difficult to time it; indeed, as a rule it will be impossible, or only possible during a more or less limited period.

110.—In the case of horizontal watches of the ordinary size in regular wear, provided the diameter of the cylinder is very small (the wheel being large) so that the friction during rest may be reduced, the cylinder escapement gives unsatisfactory results. With a ruby cylinder, where the friction is less than on metal, a watch is usually less satisfactorily regulated than with a steel cylinder, assuming both to have been recently cleaned. It often happens that in the course of one or two months or possibly more, a ruby cylinder will give fairly accurate timing, and the attainment of this result always coincides with an increase in the consistency of the oil.

111.—The double hook escapement, in which the period of rest is very brief, is seldom capable of accurate timing without the correcting influence of a fusee.

112.—The duplex escapement, when scientifically and carefully constructed, is superior in timing to the cylinder escapement; the friction during rest in the first case is nevertheless greater than in the second.

113.—Lastly, of the detached escapements ordinarily met with, the lever escapement, one of the best, is the one in which the friction is very considerable and varied; this fact is at once evident, if we add together the several resistances due to the weight of the balance, the six pivots in the escapement, the *draw* or recoil of the wheel, the pressure and friction occasioned by the passing of the tooth over the pallets, and by the entrance of the ruby pin into the fork, and finally the resistance due to the oil required by the numerous points of contact, the thickening of which in the course of time increases the several opposing forces.

Yet, notwithstanding this large amount of friction, watches with lever escapements maintain a very great regularity in their movement, that is, a regularity which abundantly satisfies the requirements of the ordinary public; but the greater number of such portable timekeepers as are provided with the detent escapement, in which friction is much less, are the bane of those watchmakers who are called upon to regulate them. At least the greater half of this class of watches, although constructed by the

most skilful workmen in the factories, vary in their rate from time to time. To secure *the best possible results* from such an escapement, we must, as already observed, provide a compensated balance (compensated, that is, in reality and not solely in name, as is the case with those worthless balances now brought into the market in such profusion), and with an isochronal spring. But such delicate work can only be accomplished by chronometer springers, first-rate workmen, rare specialists, and they alone have learnt by long experience the practice of such slow, difficult, and expensive timing, which is, in consequence, entirely out of the question in the case of ordinary pocket timekeepers.

114.—In the two succeeding paragraphs, the above article is summarised in the form of axioms, for it appeared necessary first to discuss the subject in some detail, in order to finally set at rest a question which should have been long since decided.

I.

In the case of escapements with frictional rest our aim should be to render the friction as uniform as possible, and to reduce the destructive action on the surfaces to a minimum. *The primary aim should be to determine accurately the radius of the locking surface*, since on it depends the amount of the friction as well as its correcting influence on the inequalities of the motive power.

II.

In accurate and scientific timekeepers, that is in chronometers (really deserving the name), the friction of the escapement should be reduced as much as possible, care being always taken to avoid weakening the parts and to ensure a perfect contact for a sufficient length of time. *Excessive friction entirely or in great part nullifies* the effects of isochronism and compensation.

Conclusion of the Introduction to the Study of Escapements.

115.—We will now conclude this review of the general principles to be borne in mind when designing, or even merely improving, an escapement. What has been said will suffice to throw some light on the subject, and to show that theoretical and practical knowledge is required which cannot be neglected with impunity.

We would only add a remark which we shall often have occasion to repeat; the means of attaining to regularity in machines will be best secured by looking at the subject as a

whole, by a systematic general arrangement of all the parts, by an exhaustive study of their mutual influences, and finally, by a judicious choice of expedients, and not by looking at such and such a fact or such an isolated property; in Mechanics, as in nature, goodness or badness is but a relative term.

It will be well to remember that each period in horological science has possessed its universal panacea; at one time it was isochronism; at another, perfect compensation; at a third, constant motive power; at yet another, the weight of the regulator so excessive that there was barely sufficient motive power to impel it; and so on. The remarkable phenomena here referred to have only advanced the art of Horology when, as it were, blended together, being adapted the one to the other, and modified in accordance with their mutual relationships.

It is always wise to listen to the last words experience has to say on novelties, or what are at least put forward as such, whose acceptance is too often due to ignorance or to charlatanism, which, in the absence of any merit or intellect, resorts to this convenient but well-worn method to make a pedestal for itself, where, unfortunately, we too often see it mounted at the present day.

Moreover, it must never be forgotten that in delicate mechanism, such as timekeepers, a want of care in their construction, a faulty selection of material, one fault neutralised by another, etc., points which frequently are not to be detected by the senses, suffice to explain the good results obtained for a time from arrangements radically wrong in principle, and which never fail to fall into disuse in the course of a few years. These exceptions *have always proved the rule*, confirming the truth and usefulness of the laws of Mechanics. Unfortunately, that science is only known in name by the majority of modern watchmakers, and even by a great number of those who write on their art; and this led that distinguished mechanician, M. Résal, to say with reason, "It often happens that watchmakers, who have commenced work without being properly grounded in the subject, demonstrate one day the converse of what they had the day before held to be correct. They would abridge their arguments and give a lasting character to their labours if they would only rely on the immutable laws of Physics and Mechanics, as is done by the designers of heavy machinery."

TABULAR SUMMARY

Of the General Mechanical Principles which bear on Escapements.

Symbols Employed.—C, circumference; D, diameter; S, space; F, force; $g = 32{\cdot}2$ (9·81 on the metric system), the velocity acquired by a body falling for 1 second under the influence of gravity, measured in feet (or metres) per second; H, height; I, inertia; M, mass; W, weight; P, power; R, resistance; r, radius; T, time; V, velocity.

The foot-pound (or force required to raise 1 pound weight through a space of 1 foot) is the unit of mechanical work.

116.—Weight. The specific gravity of a body is its density as compared with that of water (**47**). $W = M \times g$.

117.—The Mass of a body is given by the formula $M = \frac{W}{g}$

118.—Velocity is the space traversed by a body in a unit of time.

119.—Space is, in general, the distance travelled, without regard to the time.

120.—Relation of Force to Velocity. Motive forces or pressures, whose intensities are constant or vary but little, are proportional to the increase which they occasion in the velocity of a body. When the force is variable, we assume it to be proportional to the increments which it occasions in the velocity of a body during infinitely small equal periods of time.

121.—Vis Viva, is the effect produced by a force when it impels a body with a definite velocity. The vis viva is expressed by multiplying the mass of the body by the square of its velocity at the moment under consideration. Knowing the mass and velocity we can deduce from them the moving force.

Hence, to double the velocity, we must apply four times the force.

122.—Centrifugal Force. The tendency of a body rotating round a point to escape from that point is given by the formula, $F = \frac{W V^2}{g \times r}$

123.—Inertia. The force required to overcome inertia (or to neutralize the vis viva) increases as the square of the velocity impressed on the body. This force is expressed by the formula $I = MV^2$ and is the same as that which would be required in order to impart an equivalent vis viva to the body (**33**).

124.—The Mechanical Work given out by a Source of Power is expressed in foot-pounds by multiplying the energy or pressure exerted by the space traversed, measured in the direction of that pressure.

125.—Momentum is expressed in foot-pounds by multiplying the mass by the velocity, thus $\frac{W \times V}{g}$

126.—Falling Bodies (Latitude of Greenwich). The velocities acquired by bodies falling freely are directly proportional to the times, and the spaces passed through are proportional to the squares of the times.

At the end of each second:	1st.	2nd.	3rd.	4th.
The velocity (in feet) is	32·191	64·38	96·57	128·76
The total space traversed is.....	16·095	64·38	144·85	257·52
The space traversed during that second is...	16·095	48·28	80·47	112·66

The velocity acquired by a body in falling freely for a given period is obtained by multiplying the number of seconds in that period by 32·2.

The distance fallen, H, being known, the velocity at the end of the path is given by the formula, $V = \sqrt{64{\cdot}4 \times H}$.

If the velocity at the end of the last second is known the height is given by the expression $H = \frac{V^2}{64{\cdot}4}$

127.—Resistance of the Air increases with the extent of the surface directly opposed to its action; with the force opposed to its passing over the surface; finally,

with the velocity of the moving body, and in a proportion which is sometimes less and sometimes greater than the square of that velocity (**21**).

128.—Uniform Motion. This is the general case of the transmission of force. $S = V \times T$.

129.—Uniformly Varying Motion. The space traversed is equal to half the sum of the extreme velocities multiplied by the time occupied (in seconds).

130.—Adhesion between Oiled Surfaces. The resistance to separation is proportional to the extent of the surfaces in contact.

131.—Pressure. Pressure is proportional to the weight or force producing it.

132.—Friction. The resistance due to friction is proportional to pressure, but independent of the extent of the surfaces or the velocity of the movement.

Exception. Between the rubbing surfaces of the escapement where oil is essential, the resistance is (approximately) proportional to the extent of the surfaces in contact (**42**).

133.—Friction of Bearings. This is governed by the same laws as the friction between plane surfaces, providing the bearings are constantly lubricated by some fatty substance. Otherwise, the less the extent of the rubbing surfaces the more rapid is the wear. When these surfaces are not of sufficient extent, any excess of pressure expels the lubricating substance; there is then a destruction of the surfaces and the friction is increased.

The Larger Pivots used in horology come under the head of bearings. For pivots of an average size consult (**43**).

Exception. The *pivots* of the last mobiles offer a resistance which is taken, on the whole, to vary directly with their diameters, providing the pressure acting on them remains the same. This value of the friction, deduced mainly from the experiments of Romilly and Berthoud, can only be accepted as a mean value, for a determination of its exact amount would require fresh theoretical and experimental determinations of a very delicate and complicated nature.

134.—Inclined Plane. When P acts parallel to the plane, P is to R as the height of the plane is to its length. If P acts horizontally, P is to R as the height is to the base of the plane.

135.—Lever. P is to R in the inverse ratio of the arms of the lever, that is, of the shortest lines drawn between the fulcrum and the directions of the forces P and R (see **1034**).

136.—The Approximate Ratio of P to R in a machine can be conveniently determined by measuring the spaces traversed by the two forces; the required ratio, disregarding friction, is the inverse of these spaces.

137.—Circle. *The ratio of the diameter to the circumference* is 1 to 3·1416. If the latter number be represented by π, we have $C = \pi D$.—The *surface* of a circle is found by multiplying the square of the radius by 3·1416, or else by multiplying the circumference by half the radius. The *circumference* increases with the diameter; the *surface* as the square of the diameter.

A disc of metal, with a diameter twice that of another disc of the same metal, will weigh four times as much, providing the thickness remains constant; if all the dimensions are doubled, it will weigh eight times as much.

138.—Sphere. The *surface* of a spherical globe is obtained by multiplying the square of the diameter by 3·1416.—Its volume is equal to its surface multiplied by a third of the radius.

139.—A Ring or Rim of a balance, wheel, etc. The flat surface may be calculated by adding together the internal and external diameters, and multiplying the sum by their difference, and by the figure 0·7854. The volume is given by multiplying the surface of the ring by its thickness.

140.—Cylinder. The *curved surface* of a cylinder is obtained by multiplying the circumference of the base by the height of the cylinder. The *volume* is the product of the surface of the base into the height.

VERGE ESCAPEMENT.

CHAPTER I.

Preliminary.

141.—The verge is, as already seen in paragraph **4**, a recoil escapement, and it is gradually falling into disuse. Hence numerous correspondents have desired that the articles devoted to its consideration might be suppressed in this second edition, but we have been unwilling to comply with their request, since there are still made, chiefly in the canton of Berne, more than 300,000 verge watches annually, and, as such watches exist in very large numbers in the provinces, watchmakers will, for a long time to come, be under the necessity of repairing them.

Some authors consider it to be of German origin, others Arabian. Pierre Dubois, in his *Histoire de l'horlogerie*,* claims it as a French invention, attributing it to Pope Sylvester II., a learned and remarkable man who flourished about the year 1000. Without attempting to reconcile these varied opinions, which are not supported with much earnestness, we would only add that all we know for certain is that it was the sole escapement used in timekeepers up to the time of Huyghens (17th century).

At first it was arranged in the manner shown in figure 9, the wheel impelling two pallets on a vertical axis suspended by a cord. A metallic rod, indented like a saw, was fixed across this axis, and carried two weights, called "*regules*" or "*regulators*." These were moved towards or from the centre of rotation, according as it was desired to accelerate or check the movement of the timepiece. Such an arrangement was designated a "*folliot*" escapement.

The improvements introduced into the verge escapement in the course of last century were entirely the result of experiment; for, with the exception of so-called demonstrations of the principle, involved and inconclusive, it was not until the present day that the researches and labours of several able men resulted in a geometrical theory of the action of the verge escapement.

* P. Dubois has written works of great value as regards the history of his subject. The recent death of this elegant and fluent writer is much to be regretted; had he not passed away so suddenly, he would doubtless have eliminated from his works errors of a technical and scientific character, several of which are revivals from a past age.

Although it has now fallen into utter discredit, yet skilful practical men cannot but confess that results more satisfactory

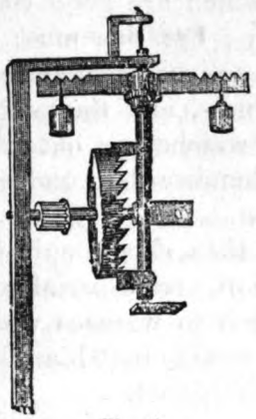

Fig. 9.

than those ordinarily met with are attainable; but such success involves careful workmanship. The very greatest care is essential in the adjustment of the relative positions of the axes, the opening of the pallets, the depth to which they are cut, the correctness of form and the equality of the teeth of the wheel, the accuracy of each drop, the proper reduction of recoil, etc.; indeed, should all these conditions be once fulfilled, they will remain so only for a short period in consequence of the wear of the pivot holes, the enlargement of which will gradually change the character of the escapement depth.

Berthoud, whose predilection for the verge escapement is well known, confesses in his *Essai* that to construct it in its highest perfection is a work of great difficulty, which can only be accomplished by very few workmen.

It will thus be seen that the production of a good verge movement involves as many difficulties as that of the majority of the best escapements used in watches; while these latter have over it this advantage among others, that by forming the pivot-holes and pallets of ruby they are rendered almost indestructible, except by accident, and the uniformity of their action is thus ensured, since the rubbing surfaces are far less liable to wear.

The excellent escapements now in use neutralise very effectually the inequalities of the motive power, a fact which the suppression of the fusee clearly proves, while with the verge a fusee carefully adjusted to the mainspring is indispensable; indeed, it often happens that on renewing the spring, it

becomes impossible to regulate the watch without first re-adjusting the fusee.

Verge watches which are good timekeepers are certainly met with occasionally; but this must rather be looked upon as due to certain actions counteracting each other than to any intrinsic goodness, and the proof is found in the fact that when once these watches get out of order, they, as a rule, remain so. No watchmaker dare undertake to restore them to their original condition.

This escapement, then, should only be employed in watches of common construction, and especially such as are thick. A watchmaker applying it to watches which are required to be accurate timekeepers, would prove himself to be utterly ignorant of the real condition of his art.

As the verge movement is so well known, it appears unnecessary to give any explanation of its mode of action.

Advantages and Disadvantages of the Verge Escapement.

142.—THE ADVANTAGES secured by using this form of escapement are the following:

1. It requires no oil on the pallets, and, as all the pivot-holes can retain a sufficient supply for themselves, the watch very rarely requires cleaning.
2. It is easily set in action (the regulating is another matter).
3. For the manufacture of the ordinary class of watches, it can be made in factories at a very moderate price.

This last fact has led one writer to say that the form of escapement we are now considering combines economy with durability, an assertion which can only be accurate if the author is referring to Franche Comté* clocks, or to the low prices charged for the repair of verge watches. There is not a watchmaker who does not know how seldom the verges of watches are found to be perfect after they have been in action for any length of time.

143.—THE DISADVANTAGES of this escapement, which have been chiefly indicated by Lepaute and Jodin, are:

1. That it requires the motive power to be always the same.
2. That it renders the watch liable to gain in its rate with

* Franche Comté was an old French province composed of the Departments of Doubs, Haute Saône and Jura. Besançon was the head town. Comté clocks, or Comtoises, are manufactured at Morez-du-Jura.—TR.

any increase in the motive power, and to lose when the reverse is the case; and this causes it to be easily influenced by changes of temperature.

3. Unless the case be thick, and there is a long axis to the verge, the oil on the lower pivot ascends to the pallet, and is transferred to the teeth of the wheel, disarranging the timing and increasing the liability to wear. If, in order to avoid this spreading of the oil, only a small quantity is applied to the pivot, it dries rapidly, the watch goes irregularly, and the pivot wears away.

4. The employment of a contrate wheel as a fourth wheel of the train is compulsory, the depth of such an one is unsatisfactory, and the friction which occurs is variable. When the axis of the balance wheel is made to pass on one side of that of the contrate wheel, these faults become still more serious.

5. The pivot of the verge wheel nearest to the verge is so strained by the reciprocating action to which it is subjected that its pivot hole rapidly becomes oval.

6. In comparison with other escapements, the balance can only traverse very small arcs without over-banking or banking.

7. The position of the verge wheel renders it compulsory that it be of small dimensions, and, the lever acting on the pallets being thus shortened, the force exerted is proportionately increased. It must be remembered that by diminishing an escape-wheel we increase both the friction and the sensitiveness of the balance to variations in the motive power.

8. The exact amount by which the pallets overlap the teeth of the wheel, that is, the amount of *lift*, is with difficulty maintained constant, since the rubbing surfaces wear away, as ruby cannot be used either for pallets or holes.

9. Lastly, far from being independent of the motive power this escapement is most closely dependent on it. The several mobiles meet when travelling in opposite directions, and this occasions impacts and friction of the most trying and destructive kind. Whenever the watch is cleaned, the escapement will require repairs, and these, if not done with skill and intelligence, will change its initial condition rendering it imperfect for a long time; possibly for ever. For, be it observed, the escapement is far from being simple, as many watch-jobbers pretend and as some authors have stated, finding it convenient to conceal, under this pretext of simplicity, their ignorance of the principles on which it acts.

DIMENSIONS IN VOGUE AT DIFFERENT EPOCHS.

144.—THIOUT.—Julien Le Roy and Sully have given in Thiout's Treatise* the proportions which their great experience led them to adopt, together with an attempt at explaining the action of the escapement. The following paragraph is extracted from their article:—

"Considerable judgment is necessary in designing the teeth of the balance wheel, as well as a great amount of skill and care in its construction. In this escapement there are three main points which require to be specially correlated, namely, the depth between the teeth of the wheel and the pallets, the form of these teeth, and the angular opening of the pallets themselves."

This paragraph is followed by a train of abstract reasoning (much of which is difficult to understand, and contains important errors) intended to prove the advantage of the following proportions, regarded by Le Roy and Sully as the most convenient mean values and the best adapted for avoiding extremes in either direction.

Inclination of the teeth to the axis of the wheel, 25° to 27°.

Opening of the pallets, between 95° and 100°.

Depth of the escapement, two-thirds of the width of the pallet.

Thickness of the pallets, half the diameter of the verge axis.

Width of the pallets, six-tenths of the space between one tooth and the next (or, more accurately, $\frac{189}{302}$).

145. — FERDINAND BERTHOUD. — "The verge escapement is the best fitted for the accurate measurement of time." (We now well know that this is not the case.)

"The escape-wheel teeth should be small and close together, so as to reduce the *drag* on the pallet, and consequently the friction.

"In order that the recoil may be reduced, the wheel should have but few teeth. Assuming the body of the verge

* Thiout, the elder, watchmaker to the Duke of Orleans, published his *Traité d'Horlogerie* in two large volumes in the year 1741. Referring to it, Moinet observes: "Thiout's work is without doubt very badly composed, and the printing is still worse; errors of orthography and punctuation often occasion contradictions; but, nevertheless, readers with a knowledge of the subject may still find useful ideas in it, and some sound advice. The many faults met with in it belong, to a great extent, to the period of its publication."

to remain the same, the lifting arcs would thus increase, and the recoil would be less in proportion; this will conduce to isochronism in the vibrations.

"Since the recoil tends to destroy the pivot holes, and thus to alter the lifting arcs, it is essential to reduce it as much as possible, and cause it to occur when the action of the tooth is almost central; otherwise, the balance will not complete its vibrations freely, and it will further be influenced by variations in the motive force.

"The body of the verge must be reduced as much as possible, in order that it may be brought close up to the wheel, for then (1) the friction is less, because the pressure of the wheel remains the same, while the drag is diminished, and (2) the lifting arcs would be greater—the supplementary arcs being consequently less; hence less irregularity will be occasioned by a varying motive force.

"The pallets should be cut short of the centre, so that the drop may be reduced."

146.—Several contradictions, such as the following, may be found in these extracts from Berthoud. He recommends wheels with few, and therefore large, teeth in order to secure isochronism; and, in another place, wheels with many, that is small, teeth so as to diminish friction, and he never attempts to reconcile these two extremes; then he advises that friction and recoil be reduced, and yet requires pallets which are cut short of the centre; it is well known that such pallets occasion more friction and recoil than any others.

It is difficult to explain the fondness which Berthoud had for the verge escapement, for all he says about large arcs of vibration, the necessity of diminishing friction, and of rendering the escapement as free as possible, etc., appears as though it had been written in favour of dead beat escapements as opposed to the verge, and his views, so favourable to this latter, are antagonistic to those of contemporary writers, notably of Lepaute and Jodin.

Berthoud in no way explains the principle of the verge escapement, if we except a few vague passages met with at rare intervals in the course of the two volumes of his *Essai*. Three points are clearly demonstrated by this author, (1) the pallet should not overlap the teeth by more than two-thirds of its own width; (2) the friction, especially that due to recoil, is

the main objection to the escapement; and (3) the entire oscillation of the balance rarely exceeds a half circumference.

Opening of the pallets 90°, 95°, and even 100° when it is desired to increase the vibrations and at the same time to avoid *over-banking* or *banking*.

Inclination of the teeth of the wheel 15° to 20°. Occasionally it is as much as 25°.

The total arc of vibration should be three times the lifting arc.

147.—TAVAN (See a Memoir published by the Geneva Society for the Advancement of the Arts). Opening of the pallets, 100°.

Inclination of the teeth, 25°; lifting arc, 40°; entire amount of vibration without over-banking, 220°.

The width of the pallets measured from the centre of the axis should be $\frac{2}{11}$ths of the diameter of an 11-toothed wheel, $\frac{2}{13}$ths that of a 13-toothed wheel, and $\frac{2}{15}$ths with 15 teeth.

The depth should be so adjusted that when the opening of the pallets is 100°, and these occupy the positions indicated in figure 10, page 64, the tooth shall cause each pallet to traverse an angular path of 20° (thus completing the entire lift of 40°). The points of the teeth will then be in the vertical plane indicated by the dotted line MN.

"With such a depth the requisite amount of drop is secured without any catching; this, then, is considered in practice to give the best results."

It will be evident from a simple inspection of the figure that the requisite depth is attained when the teeth and pallets overlap by about two-thirds of the width of the latter.

148.—MOINET (In an article by Duchemin). Opening of the pallets from 100° to 110°; it may be as much as 115°.

Width of the pallets, measured from the centre of the axis, half the interval between one tooth and the next.

Total lift 40°; inclination of the teeth from 30° to 35°.

"The modern practice of cutting verges tends towards Berthoud's principle, since it allows of the wheel being brought into closer proximity to it; but this must not be carried too far, for *the lever being thus shortened, a balance less heavy, and therefore less able to overcome the clogging of the oil must be employed.*"

149.—M. WAGNER (In a Memoir on Simple Escapements). Opening of the pallets from 100° to 115°; lifting arc, 50°; entire arc of vibration, 170°.

The other proportions are the same as those adopted by Duchemin.

"If," says M. Wagner, "the rules laid down by this horologist be followed, the rate of timekeeping of watches with the verge escapement will equal that of watches of the horizontal construction." (Experience has shown that the reverse is the case.)

"The length and opening of the pallets should vary with the angle of oscillation which it is desired the balance or pendulum shall describe, not with the diameter of the wheel as has hitherto been assumed; at the same time the distance between the teeth is a material factor in such an adjustment."

"Notwithstanding all that has been said against this escapement, and the lightness of the pendulum usually employed (he is here speaking of marqueterie timepieces and comtoise clocks), the uniformity in the rate of a considerable number of such timekeepers is quite as satisfactory as when the modern escapements, so much vaunted, are employed." (The opening of the verge pallets is relatively very *small* in them.)

"The notion that this escapement cannot be employed in conjunction with pendulums as heavy as those impelled by other escapements is erroneous."

"The main object held in view in deciding upon the foregoing conditions has been to obtain the greatest effect *with the least possible amount of friction.*"

"I would observe that in order to reduce the friction to a minimum, it is essential that the action of the tooth on the pallet during the entire oscillation, especially during the supplementary arc, should occur as nearly as possible on the line joining the centres of the axes of the pallets and the wheel."

"The action of the tooth against the pallet has a very slight effect on the friction of the pivot." (This is only true in the case of timepieces in which the angle of the verge pallets is small, for with watches the converse is daily proved to be true by the rapid enlarging of the pivot-holes.)

At the conclusion of a comparison of the proportions recommended by Le Roy and Wagner, we read:—

"In the first, where the opening of the pallets is considerable, there will be an increased friction both at the points of the teeth and at the pivots of the verge wheel in consequence of the oblique action of the pallets on the teeth during the recoil of each tooth, or during the supplementary arc. *It is*

thus demonstrated that the more the opening of the pallets is increased beyond the extent necessary to avoid over-banking, the more friction occurs, and, as a consequence, the greater will be the variation in the rate of timekeeping.

"It has already been shown that the variation due to friction is directly proportional to it."

"The obliquity of the faces of the teeth should increase as we increase the arc of vibration, the inclination of the faces being a few degrees greater than one-half of the supplementary arc" (taken only on one side).

"I must here remark that the friction of the point of the tooth on the face of the pallet increases with the arc of vibration, a circumstance which cannot be prevented in this arrangement. *This fact proves that the escapement is all the more capable of giving accurate results as the oscillations are made shorter.*"

150.—After such quotations as the above, it is difficult to understand how the author we are considering can have preferred, in the case of a watch, to adopt the proportions recommended by Duchemin. It is certain that he, a skilful watchmaker, is prejudiced, throughout his work, by a consideration of various facts observed in clocks, and especially in the larger class of timepieces, in which this form of escapement works in connection with a pendulum. Thus, for example, he does not attach sufficient importance to the *recoil*, which, while quite insignificant in the escapement of a clock, where the entire oscillation is of small extent, becomes a cause of wear and very serious irregularity in watches where the amplitude of the arc described is about eight times as great as in clocks.

We will confine our criticism to the above remarks, and this reserve in discussing the views of a horologist now living will be easily understood.

Although quite unable to explain M. Wagner's choice of the proportions suggested by Duchemin, we must here record that it is to him, so well known by the beautiful turret clocks of his manufacture, that we owe the geometrical theory of the escapement now under discussion.

Table showing the Proportions recommended by various Authors.

	Inclination of the Teeth of the Wheel.	
Le Roy and Sully.....	25° to 27°	Minimum 15°, Maximum 35°.
Berthoud	15° to 20° then to 25°	
Tavan	25°	
Duchemin-Wagner	30° to 35°	

Opening of the Verge Pallets.		
Le Roy and Sully	95° to 100°	Minimum 90°, Maximum 115°.
Berthoud	90° to 95° & even 100°	
Tavan	100°	
Duchemin-Wagner	100° to 115°	

Two remarks in connection with this Table.

151.—On examining the above table it will be seen that the earlier proportions differ from the more modern in this: with the older watchmakers the opening of the pallets varied between the limits 90° and 100°, and with recent makers the limits were 100° and 115°. The maximum opening in the former case is exactly the minimum in the latter. It seems strange that it did not occur to such horologists as Le Roy, Sully, Berthoud, Jodin, etc., who occupied themselves with the vertical escapement during many years, that the addition of a few degrees to the opening of the pallets would ensure a greater degree of accuracy than that to which they had then attained.

A prolonged experience of this escapement had led them to discover the difficulty which the advocates of a considerable opening, carried away by the system of very large oscillations at one time in favour, had not foreseen, but which they could not have failed to discover after a few years passed in watch-jobbing.

It is a fact equally worthy of remark that whenever practical men have found fault with Le Roy for making the opening of the pallets in his verge clocks too great and the pallets themselves very short, thus rendering a light balance and large oscillations essential and introducing excessive friction, they have condemned the modern system of escapement with very open pallets, which are also deserving of the above criticisms, and not less justly so than the clock escapement of Le Roy.

CHAPTER II.

PRINCIPLE OF THE VERGE ESCAPEMENT.

Tangential Escaping.

152. *Lift.*—Let *a b* (fig. 10) be the line which passes through the centres of movement of the wheel and verge; z y the line of action of the motive force, and, at the same time, the projection of the plane passing through the points of the teeth. It will be at once seen, from a simple examination of the figure, that the only case in which the escapement acts tangentially is

when the face of the pallet is in the plane *a b*. For the line, Z Y, indicating the line of action of the force, is then perpendicular to the radius *a q* and consequently is a tangent to the circle *p q r*.

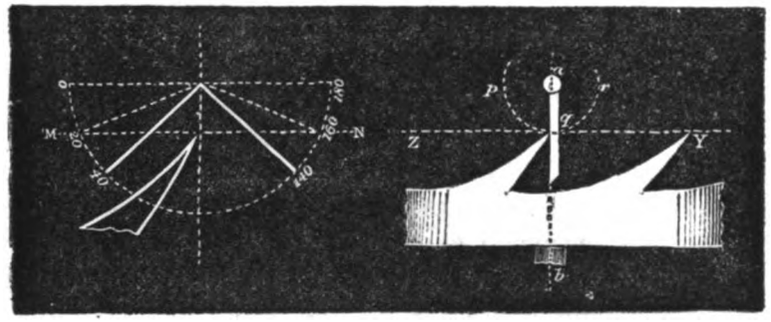

Fig. 10.

Recoil takes place when the pallet, having received the impulse from the tooth, compels this latter to move in a reverse direction. But, since the drop only occurs when this tooth has passed the line of centres, it follows that, while the tooth is forced backwards, friction is brought into play which diminishes as the action takes place nearer and nearer to the line *a b*.

153.—Hence: *In the verge escapement, the friction introduced, the resolution of the motive force and the pressure on the axes, during the lift and recoil, will increase as the action takes place farther from the line of centres, thus becoming more and more oblique to that line.*

The *lift* is accompanied by disengaging friction. The *recoil*, on the contrary, is influenced by engaging friction; *it is in the recoil, therefore, that the primary fault of this mechanism consists.*

The pallet escapements of clocks are so constructed that the action shall take place much less obliquely to the line of centres than in watch escapements; thus it is that the former continue to work satisfactorily for a much longer period than the latter.

To Design a Verge Escapement.

154.—The method adopted in preparing such a design is based on the principle laid down by M. Wagner that "The length and opening of the pallet should vary with the arc of oscillation it is desired the balance or pendulum shall perform."

Consider the case of a clock escapement which is required to describe 8° of lift and 6° of supplementary arc.

The opening of the pallets will therefore be 14°.

Draw A B (fig. 11) the line of centres; at equal distances on

either side of this line draw two lines H and X parallel to it, such that the distance H X is equal to the interval between the points of two teeth, an interval which, if not already known, must be first determined.

Through the point A, taken as the centre of rotation of the axis of the verge, draw the line F G, perpendicular to A B, and set out the angle O A Y, equal to 3° (half the supplementary arc); then the angle C A Y, equal to 3° + 8° or 11°. The entire angle O A C is therefore 14°.

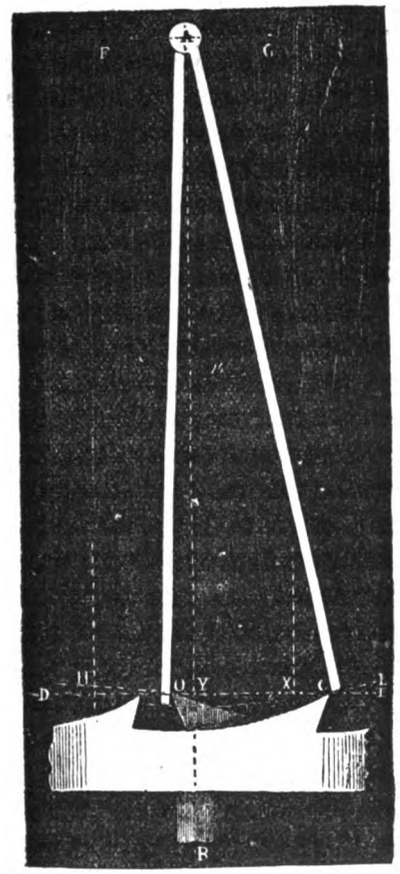

Fig. 11.

Now, by means of a scale moving parallel to the line F G, determine the point along the line A C, at which O Y is equal to X C. The point C thus fixed upon will be the extremity of the pallet A C, and will indicate the position it occupies at the end of the lift; that is, when the tooth is just on the point of escaping.

The line E D drawn through C, perpendicular to A B, fixes the plane in which the teeth of the wheel rotate.

From the point A as a centre, and with the distance A C, describe an arc of a circle in order to ascertain the length of the other pallet. As to the position of the tooth which is caused to recoil immediately on the commencement of the supplementary arc, it is determined by the intersection of the lines A O and E D.

The Opening.

155.—Figure 11 represents a verge with an opening of 14°. It shows that the motive force acts on the tooth O, in the direction E D, and the recoil acts in the direction H *i*; that, therefore, the force causing the wheel to go backwards acts nearly perpendicular to the line of centres A B in which the axis of the wheel lies. The action takes place in opposition to the motion of the wheel, so that it travels a few degrees backwards. This recoil occasions no difficulty, since the play in the depths is sufficient to prevent its being seriously resisted on the part of the wheel. Moreover, the point O is so close to the line of centres A B, that the pressure on the pivots is reduced to a minimum, and the action of the tooth on the pallet is not resisted by any considerable friction; hence, in such clock escapements as have a moderate opening, very little wear of the surface of the pallets occurs where this recoil takes place.

In ascertaining the direction of the forces during the lift, we observe that the line K L (fig. 11) is much more inclined than H *i* to the line D E, along which the motive force acts, and that the point of the tooth acts much farther from the line of centres. But since the friction during lift is less intense than that during recoil (the first being disengaging and the second engaging friction), the escapement is, notwithstanding these facts, well adapted to resist wear.

As an example of this form of escapement, we may quote that found in Comté clocks. It wears much less rapidly than in a watch, and will continue in action maintaining a fairly uniform rate of time-keeping for five or six years or even longer.

Let us now consider the right-hand diagram in fig. 12, where the opening of the verge is as much as 115°, and the directions of the several forces are indicated by D E, H *i*, K L (the same letters being used as in the former case); it will be observed that, as we increase the angle of the pallets, the angles which H *i* and K L make with the line of centres A B become more and more acute. From this it follows that the

force acts more obliquely, and the resistance tends to act in the direction of the axis of the wheel; a very unfortunate circumstance, for an increased proportion of the impelling force has to be resisted by the point of a pivot resting against a rigid plate. The wheel, held between the point of a tooth and its lower pivot, acts as if it were an inflexible rod pressed at its two ends. The pressure on the axes is very great, as is also the extent to which the force is resolved; the motive power is partially paralysed, and, owing to the inertia of the balance, the opposing forces produce such an amount of friction that the pallets, however hardened, are unable to resist its action.

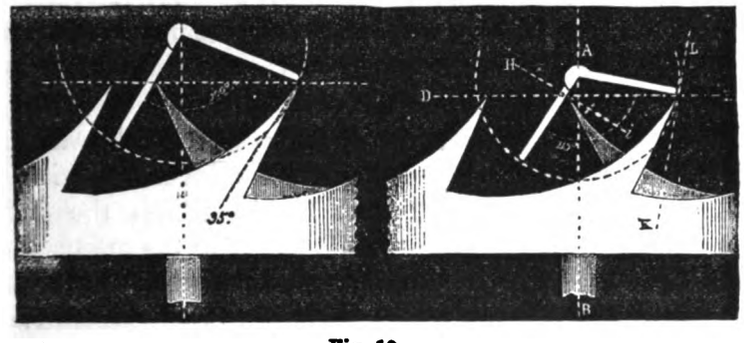

Fig. 12.

We thus see, as M. Wagner has pointed out, "that the vertical escapement is all the more capable of giving accurate results as the oscillations are shorter." And, since the arc described by the balance and the opening of the verge are correlated, we may add: a watch escapement will be the more lasting if the angle of the pallets is no greater than that found to be absolutely necessary and justified by experiment.

As will be shown in the sequel, it has been fixed at 100°.

The Lift.

156.—It has been already mentioned that the extent to which the pallets are planted in the teeth of the balance wheel is not a matter of indifference; it depends on the angle of these pallets. The same may be said with regard to the distance between the points of the teeth and the axis of the verge: it must be diminished as we increase the opening of the pallets. With an angle of 40° (fig. 13) the pitch will be from one-seventh to one-sixth of the width of the pallet measured from its edge. The space between the teeth and the body of the verge will be rather more than the distance apart of the points of two successive teeth.

Assuming the opening to be 100°, the pitching will be about two-thirds of the width of the pallet. The interval between the points of the teeth and the verge axis will be from one-fourth to one-sixth of the distance apart of two teeth (fig. 12).

Finally, with an angle of 115° the pitching will be from six-sevenths to five-sixths, and, in practice, this is expressed by saying that the wheel falls just against the body of the verge. In this case the distance of the points of the teeth from the axis of the balance is about one-tenth of the interval between two teeth.

The above figures, although only approximate, are quite sufficient. Exact figures would be useless, and could not be accurately adhered to in the escapement we are now considering.

157.—These remarks on the amount of pitch are equally applicable, if we reverse the ratios, in discussing the width of the pallets, which should vary inversely with their opening. (See figs. 11, 12, and 13, in which the interval between the teeth is constant.) The greater the angle, the less the width of the pallets; thus the wheel acts on a shorter and shorter arm, the impulse communicated to the balance varying with the length of this arm and the period during which the motive force acts on it.

It therefore becomes evident that some relation must exist between the motive force, the length of arm above referred to, and the weight and diameter of the balance; but a problem of this complexity is beyond the range of calculation, and must be decided, for the present, by experience. The best horologists have determined upon about two-thirds of the width of the pallet as the extreme limit of pitch permissible in an ordinary watch, and, it may be observed, this amount is exactly that which corresponds to an opening of 100°.

The Supplementary Arc.

158.—Every watchmaker is aware that in the case of most of the escapements met with in practice, it may be laid down as a general rule, that the watch maintains its going better according as the supplementary arc is greater in comparison with the lifting arc (**95**).

With a lifting angle of 40° we obtain, as a rule, from a verge escapement a total vibration of nearly 180°. If the lift

be increased beyond 40° to, say, 50°, the entire oscillation is not increased in the ratio of 40 to 50, so as to give an arc of 225° (nearly three-quarters of a complete rotation), as this proportion would indicate; the sole effect is that the vibrations are characterized by taking place rapidly and irregularly. This comparative decrease in the supplementary arc when the entire

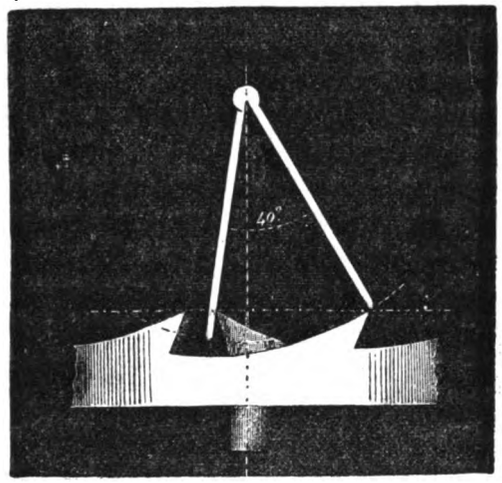

Fig. 13.

oscillation exceeds 180°, to which, to the best of our knowledge, attention has never before been directed, is explicable if it be remembered that: (1) the resistance owing to recoil is considerably increased; (2) the lever arm is shortened, thus causing less force to be applied to the balance by the wheel; and (3) the edge of the pallet rubs against the face of the tooth, so reducing the supplementary arc. This banking occurs towards the end of the oscillation when it is only 170°, and becomes the more serious as this amount is exceeded. The resistance occasioned by the banking is such that, in a watch liable to rough wear, it causes the larger oscillations to be more rapid than the small ones, a circumstance with which all watchmakers are acquainted.

The conclusion we arrive at is, that whenever the amount of lift exceeds that necessary to produce a balance vibration of the requisite amplitude, rapid destruction and irregularity cannot fail to occur in the mechanism. Since long experience has demonstrated 40° of lift to be amply sufficient in watches, it will be well to adhere to this number, especially as it has, besides, the advantage of securing *the greatest relative*

amount of additional arc; but, as it has been shown that the amount of lift and of opening are mutually dependent, it follows that the best opening is that which secures a lift of 40°, that is an angle of 100°.

Recoil.

159.—The period of recoil amounts to one-half of the entire additional arc. In the escapement of an ordinary clock, with a complete oscillation of about 20°, the recoil will occupy 6°, while in a watch with a vibration of 180°, the recoil occupies 70°. But these two numbers only express the relative extent of the recoil in the two cases, and cannot be taken as proportional to its intensity; for, on such a supposition, the disproportion between the two examples would be all the more marked. This fact will be easily understood if we remember that, as has already been pointed out (**155**), the thrust of the balance, the cause of recoil, approximates towards a direction parallel to the axis of the wheel as the angle of the pallets is increased. Hence it is evident that rules which apply in the case of a clock escapement, must be very considerably modified when we come to apply them to a watch.

Bankings, which are in great part avoided by increasing the opening, are less objectionable than recoil.

The recoil is the grand defect of the vertical escapement as arranged in a watch, and this conclusion has been arrived at by nearly every practical man, even Duchemin himself having ultimately recognized it. (See *L'Art de conduire les pendules*, etc., by M. Robert, page 248.)

Besides interfering with the due performance of the supplementary arc, the friction, occurring as it does at two distinct periods of the vibration, in nearly every case causes the pallets to wear, and occasionally these are cut through if the watch has been long enough in action. It is always at the point corresponding to the commencement of recoil that the wear of the pallet first occurs, and this is very conspicuous through an eye-glass or even to the naked eye. As soon as a pallet is at all worn, the points of the teeth become distorted, the steel wears rapidly over all its rubbing surface, and the variations in time-keeping are more and more marked. The energy of the recoil is, moreover, indicated by the rapid enlargement of the pivot-holes, and frequently by the wear of the pivots themselves.

All attempts to render more lasting the several parts of this escapement, such as a gold wheel, a hardened steel wheel working with a small quantity of oil on the points, ruby pivot-holes, etc., have utterly failed, from the simple reason that none of them get rid of recoil, the radical defect of the escapement.

It follows from what has been said that, since the extent and harshness of the recoil increase with the angle of the pallets, it is advisable to adopt as small an opening as possible if we would reduce the sources of wear and irregularity which, in any case, will show themselves soon enough. As has been already seen, there are many good reasons for not exceeding, or, if at all, by very little, 100° for this opening of the pallets.

Inclination of the Teeth of the Wheel.

160.—The inclination of the teeth of the wheel must clearly be as great as possible in order to prevent the pallet striking against their faces. Modern horologists make it 35°; but this seems to be a mistake, for, since it is necessary to cut away a considerable portion of the back of the tooth in order to avoid catching, a thin elongated tooth will be produced with so little rigidity that great difficulty will be experienced in securing and maintaining trueness in the wheel.

The left-hand diagram of fig. 12, page 67, in which the several parts are represented to scale, does not require explanation.

It will be well not to exceed 30°. This amount of inclination allows of the teeth being sufficiently rigid, and does not require, to nearly the same extent as 35°, an amount of care and delicacy which it would be ridiculous to expect of the watch-jobbers who have to do with this second-rate class of watch. It must, nevertheless, be confessed, that the only objections to an inclination of 35° are the difficulty of cutting the teeth and of maintaining the wheel true.

RECAPITULATION.

161.—It is shown in the preceding articles that the opening of the pallets and their pitch with the teeth, the lift, supplementary arc, entire vibration, the amount of recoil, etc., are all mutually related, and not one can be treated independently; it only remains, after choosing a definite basis, to fix the relative proportions of the several parts.

Now as soon as it has been shown that if a total vibration

of 180° is exceeded, the rate of wear of the working surfaces, as well as the causes of irregularity in going, increase rapidly (**158**), we at once have an exact datum to guide us in the construction of the escapement. That is to say, the motive force, weight of balance, lift, friction, etc., are all secondary to this arc of vibration, and must be so co-ordinated as to produce it under the best possible mechanical conditions.

And, in conclusion, it must be borne in mind that we have already shown the adoption of 100° for the angle of the pallets to secure, in addition to an amount of lift of about 40°: (1) an entire vibration of sufficient extent (for it may be as much as 180°) which will be completed with much greater freedom than if more extended; (2) the most free and, at the same time, the greatest additional arc it is possible to have in comparison with a given lift; and (3) an overlapping of two-thirds the width of the pallet, which gives neither so much leverage as that the balance loses all control over the wheel, nor so little as to occasion a *setting*, an infallible sign that the motive force impelling the pallet is insufficient. All these well-known facts, proved both by theory and experience (see the works of Le Roy, Sully, Berthoud, Jodin, Lepaute, Tavan, Perron, Duchemin, etc., etc.), are amply sufficient to demonstrate that we must in no case exceed 100°, and when it is remembered that three or four extra degrees will cause not only the pitch, but also the energy of recoil (already so considerable and prejudicial at 100°) to increase in a rapid ratio, it is surprising that the amount of opening should still be considered open to discussion; and one is astonished at the oft repetition of such a proposal as to employ a lift of 50° to produce a total oscillation of 170°, when the experience of over a century has proved such an oscillation to be easily obtainable with a lift of about 35°.

It should not be forgotten that the extent of oscillation is a *consequence* of the greater or less freedom of the train, the accurate adjustment of the contrate wheel depth, the weight of the balance, the uniform action of the balance-spring, and of the several frictions in the escapement.

We have as yet made no reference to the drop, or to the exact width of the pallets, but they will be discussed further on. It should be observed, however, that Duchemin gives only one width for the pallets, and yet it certainly must vary considerably if the opening is increased from 100° to 115°.

Table showing the several dimensions of the Escapement in Verge Watches of the present Day.

162.

Opening of the pallets	100°
Total lifting arc	40°
Inclination of the teeth to the axis of the wheel	30°
Maximum complete oscillation	180°
Oscillation possible without overbanking	220°

Width of the pallets (measured from the centre of the axis), a little more than half the distance between the points of two successive teeth, or approximately:

With a wheel of 11 teeth $\frac{1}{5}$th (bare) of its diameter.
" " 13 " $\frac{1}{6}$th " "
" " 15 " $\frac{1}{7}$th " "

The proportions should be finally adjusted after the lift and drop have been verified with all the parts in position.

If it be required to slightly reduce the angle of the pallets, we must somewhat increase their width, decrease the extent of lift and of the complete oscillation, and make the size and weight of the balance greater, taking care that these changes are in due proportion.

The teeth should be so cut away behind as to avoid any catching of the edge of the pallet on them.

163.—IN COMTÉ CLOCKS:

The lifting arc is about 2° or 3° on either side.

The supplementary arc is a few degrees greater; so that the complete oscillation is between 10° and 15°.—The angle of the pallets is between 50° and 60° (**936**).

These pallets are at times flat and of the same thickness as the body of the verge, whence it arises that the angle formed at the axis is somewhat less than the actual angle of the pallets. As to those in which the face is concave, they act very nearly as though it were a plane passing through the axis of rotation.

As these escapements have a more open angle, and narrower pallets than are theoretically correct, a very considerable amount of friction is occasioned. The choice of such proportions by the manufacturers, however, is due to the fact that, when so disposed, the escapement requires a less weight on the pendulum, less accurate adjustment in the workings, and admits of greater supplementary arcs, than if constructed in the manner which theory would indicate to be the best.

We have already treated the verge escapement in a sufficiently exhaustive manner and, prior to it, the question of friction, to render useless any further particulars.

PRACTICAL DETAILS

Bearing on the conditions above laid down.

164.—Nearly, if not quite, all the best practical horologists are agreed on this point, that 40° of lift is amply sufficient, and that the only effect of exceeding this is to introduce fresh causes of wear and irregularity.

In watch factories, where during a long period experiments were made on the subject of this angle, 100° has been fixed for forty years past as a maximum opening.

When the balance of a watch has what is called a *falling off in the crossings*, it is generally sufficient, providing the pallets are wide enough, to slightly close the verge opening in the flame of a lamp, in the manner well known to watch-jobbers, by which a greater liveliness in the motion and a more uniform timing will be secured, if the balance be not too light.

If a competent watchmaker be asked what is the chief characteristic of too great an opening, he will unhesitatingly reply that, in the absence of any other means, he would detect it by the balance vibrating to its full extent so long as the oil is fresh and the rubbing surfaces retain their initial state of perfect polish, but falling off in the crossings after going for a few weeks and constantly varying after that. Now what he regards as a too open verge is one of which the angle exceeds 100°, as we have repeatedly proved to be the case, and hence arises that expression commonly employed in the trade: *the verge must be opened rather more than a right angle.*

Old watches were capable of maintaining their rate for about three years. Modern ones, while favourably constructed as regards thickness, go with difficulty for eighteen months or two years, and at the end of this period their pallets are nearly always much worn. Such watches usually go very well during the first few months, and very badly during the remaining period.

Finally, all these facts, so crucial and so well known, are confirmed by the unmistakeable evidence of English watches. The English system is utterly different from that practised by us; the pallets are less open, the balance is heavier, and the complete vibration less, all objectionable features in the eyes of those who recommend the large opening. And yet such an escapement gives a rate equal to that of recently cleaned French watches with widely open verges, but with this

important difference: that those of our manufacture soon go irregularly and become worn, while the English timekeepers maintain their rate for years, and even then their rubbing surfaces are only slightly worn, if at all.

One proof, and that unanswerable, of the superiority of this latter class of escapements is the excellence of their rate of timekeeping, that is, within the limits which are reasonable for such an escapement, with jewelled holes for the balance pivots, an improvement not yet introduced into French watches.

The English have shown their discretion; they have endeavoured to diminish the combined effects of recoil and the banking of the pallet, and have been successful, while we have all the time been tending in the reverse direction.

Too Small—Too Great Opening.

165.—When the verge is too much closed, banking occurs, though the oscillation be very short. If the pallets have great width, the tooth acts on a lever which is too long; its action on the balance is proportionately excessive, and this latter, losing its control over the motive force, is affected by its want of uniformity.

When the opening of the verge is great, the wheel is set very near to the body of the verge. The lever on which the teeth act becomes so short and so much inclined to the direction in which the force is applied that, during the greater part of its action, the impulse is insufficient to overcome the resistance occasioned by the inertia of the balance and the thickening of the oil; the consequence is that the escapement *sets*, even with a light balance.

166.—Besides this, a reduction in the width of the pallets renders the friction more variable, because the play of the pivots and consequent increase of the pivot-holes becomes greater. Let there be two verge pallets and let each be reduced by the same amount, A B (fig. 14); it is evident that, since the friction has been transferred from A to B, it will, with the short lever, vary in the proportion of two to one, and in the second case only to the extent of one-fifth.

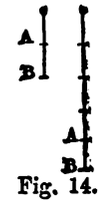

Fig. 14.

In general, when the angle of the pallets is not sufficiently large the vibrations are short and rapid, and in the converse case they soon become sluggish; either circumstance has the effect of making the watch vary in its rate.

The above essentially practical results are indeed apparent to the eye of an observer when the opening of the pallets is below 95°, or above 100°. If it be remembered that the intensity of the friction during recoil and the pressure of the edge of the pallet against the faces of the teeth become more severe as this latter amount is exceeded, and also bearing in mind that when the opening is considerable the adjustment of the escapement demands an amount of care and delicacy quite incompatible with the cheapness of the class of watch to which it is applied, it must be admitted that it should in no case exceed 100°. To attempt to force workmen to devote such care to the verge escapement as is requisite in the case of the higher class of escapements, is an error we will only indicate, it is not worth refuting.

CHAPTER III.

MANIPULATIVE DETAILS OF CONSTRUCTION.

The Verge.

167.—The verge, made of steel of the very best quality, should be carefully hardened, its pivots being tempered to a blue or rather violet tint, but its pallets only to a yellow, and the whole should be perfectly polished. Every watchmaker is aware that a verge is straightened by striking with the sharp end of a hammer head on the concave side of the body, while the convex side is supported in a horizontal position, and that it is as well not to attempt to remove the marks left by the hammer. In soldering on the collet, care is necessary to avoid over-heating of the steel, as this renders it rotten, brittle, and much more liable to wear at the points of contact with the wheel.

The verge pallets should be cut down to the middle of the body. Berthoud cut them down rather less than this in order to diminish the drop; but such a practice is inconvenient in several ways; the increased recoil accompanied by a too short drop frequently causes stoppage; moreover, as the angle of the pallets appears somewhat greater than it is in reality, there is some difficulty in determining whether it is properly adjusted.

With a verge in which the pallets are cut down beyond the centre of the axis, the drop becomes considerable. As a general rule the watch soon falls off in its rate (**170**.)

The authors quoted above do not recommend precisely the same dimensions for the pallets, but this is unimportant. It is best to make their width rather more than half the distance between the points of two successive teeth (**162**); and they can be carefully reduced until the lift and drop are properly adjusted.

The width is measured from the edge of the pallet to the middle of the body, and it is always in this sense that we have referred to the width of the pallets. It will be seen from this explanation that if the entire width of a pallet be measured it will be necessary to deduct from it half the thickness of the verge body in order to ascertain the true width of the pallet measured from the centre of rotation; and this is the only exact measure, since the total width varies with the thickness of the body.

The curvature of the outer edge of the pallet should be concentric with the axis, as though it were a radial slice cut out of a cylindrical rod. Such an arrangement has this advantage: the width of the pallets will not be reduced when it is necessary to re-polish them.

168.—The pallets must be highly polished; but care should be taken that the rouge on the zinc or copper polisher be not too dry, and it is well to complete the process with a soft piece of wood charged with the finest quality of rouge employed.

Some workmen use nothing but oilstone dust and iron for narrowing the pallets, which should be of absolutely equal width; this operation should always be completed with rouge, and the edges should be carefully rounded, for the use of oilstone powder leaves a kind of wire edge which is very detrimental both to the teeth of the balance wheel and the pallets themselves.

Soap and water is a good and convenient method of cleaning a verge, and is the one practised by the best workmen.

Verges with ruby pallets.—Verges cut beyond the centre.

169.—Watchmakers who think they have introduced a novel improvement by constructing the verge pallets of ruby are frequently met with, more especially in the provinces. They eagerly announce their *discovery* with a great flourish of trumpets, but after a few months nothing more is heard of it.

Some watches provided with ruby pallets have been known to go fairly well; and the marine chronometers by Harrison and Larcum-Kendal may be referred to as instances. But their

success depends on the correct solution of a problem in friction, and this question, though it appears simple at first sight, is extremely complex and would require for its consideration all the resources of modern science; in other words it is out of the power of watchmakers of the present day, except perhaps some half-dozen; and, as to science, it has more important work to do than to devote its attention to the verge escapement (**177**).

170.—When the verge is cut short of the centre or beyond it, it is quite possible to regulate the watch. We only need find out a certain relation and fix it. When the opening is beyond the centre, however, this relation is maintained with the more difficulty, the angle of the pallets is not so exactly determined, and as soon as the verge wears it becomes useless. We do not dwell upon these *novelties*, often attempted and as often abandoned.

The Balance Wheel.

171.—The balance wheel, formed of perfectly hard brass, should, as already shown, have as great a diameter as possible. The teeth must be of a good average thickness, so that the surface, slightly rounded at the edge which acts on the pallets, may be, if anything, somewhat thick (**41**).

When the tooth is thin the force communicated is the same as with a thick tooth, but since it is distributed over a less surface, it acts with greater energy at each point. The roughnesses of each metal are forced more deeply into the other, and the pallet is cut more rapidly, the friction continually increasing.

The same reasoning applies to pivots, which are subjected to excessive friction and wear badly when working in short metal or jewel holes, and to the destruction of a knife-edge suspension. Since the knife-edge never bears accurately throughout its entire length, there is only a series of small surfaces in contact at any instant, and they rapidly become dull.

A balance wheel must have an odd number of teeth, so that while a tooth is acting on one pallet the other pallet may always be in a gap.

An experimentalist (Perron) made, more than half a century ago, trials with balance wheels in steel and gold. These latter caused the verge pallets to wear, but those of steel gave good results when oil was applied to the pallets.

It is difficult to cut a brass wheel perfectly true unless only a small quantity is removed at a time, a thoroughly good

circular file or cutter being employed, and the formation of the teeth being accomplished by oft repeated operations; as the metal yields irregularly under the pressure of the cutter, the wheel is always found to be distorted when the ring of teeth is complete.

Care must be taken in the *dressing* of the wheel as well as in the removal of the burr from the points of the teeth and the *equalizing* of them. A wheel which does not turn true, or is unevenly divided, or turns out of flat, occasions a waste of part of the lift, drops of unequal extent, and, in short, renders the timing of the watch utterly impossible.

172.—It is a good practice to slightly corrode the teeth with nitric acid, as explained in the article on *Wear of the Pallets* (Chapter IV). But after this operation some watchmakers clean the wheel by brushing it lightly with rottenstone, burnt hartshorn, or fine charcoal, for just such a period as suffices for the cleansing of the piece.

Satisfactory results are obtained by employing a tin polisher and soft water of Ayr stone mixed with oil for removing from the teeth the streaks left by the cutter; they are then finished by a piece of soft wood with fine charcoal and oil followed by a brush dipped in similar charcoal. The use of a burnisher, which in any case must be very clean and highly polished, does more harm than good if the wheel has not been previously treated with acid or perfectly lapped, and at times it only serves to press the foreign bodies resting on the surface of the tooth into its substance.

These latter details will be supplemented by various practical directions in the following chapter.

To make a Balance Wheel True.

173.—All the tools employed for making a balance wheel true are sold by the watch-tool and material dealers; but although the price is considerable the operation performed by their means is very tedious, and the result is not always satisfactory. In place of them we may with advantage adopt the following method of M. Noriet, of Tours, a method which has been practised successfully for over twenty-five years by M. Brisbart, of Paris, and by the many pupils of that excellent teacher.

The burr having been carefully removed from the teeth, the wheel is placed in position, the escapement carefully adjusted (without play), and the dovetail is then caused to

traverse a very short space. The movement given should be no more than is necessary to cause some teeth to catch on the nose of the potence.

This catching may be produced by a pressure of the finger, and a slight play should be given to the wheel in the direction of its axis, so that the tooth may pass the pallet during its recoil.

The dovetail having been thus placed so that some of the teeth can only be caused to pass by applying pressure, and the plate being held in the left hand (one finger of which imparts motion to the wheel), as soon as a tooth catches the two following teeth must be impelled past the pallet; the catching tooth is thus made accessible, and it is easy, with a fine new rounding-up file, to reduce the inclined face of this tooth in such a manner as to prevent its catching. This can be done without danger to the pivot, for the wheel is held by the finger which presses the shoulder of the pivot against the nose of the dovetail, and the action of the file tends to give a sort of recoil movement to the wheel rather than to strain its pivots. After a little practice, this operation will be found to be neither difficult nor dangerous.

When a complete rotation of the wheel has been made, and the several teeth in which catching was detected have been adjusted, the dovetail is slightly moved again, the above operation is repeated, and so on until all the teeth pass without drop on the dovetail, etc., etc.

When the irregularities are very considerable, the methods described in Chapter IV. must be first resorted to. (Badly formed or irregular teeth.)

Pivots and Pivot-holes.

174.—If the body of the verge be not thicker than is necessary for its due solidity, the diameter of the pivots, which must be hard, cylindrical, and carefully polished, may be a quarter of the width of the pallets; experience has proved that this is the most convenient proportion for the diameter of the pivot to bear to the width of the pallet. The pivots should have a length equal to three times their diameter; their ends must not be rounded but flat, the angles being slightly rounded off, in order that the total amount of friction may be, as nearly as possible, equal in the vertical and horizontal positions, that is, in order to regulate for position.

The balance wheel pivots may have the same thickness as

those of the verge itself. They may be even finer without inconvenience; but care should be taken to make them of sufficient length, at least twice their diameter, and more if possible.

175.—The pivot-holes must be made in brass of good quality, and not too thin, otherwise they will cause the pivots to wear away. The reason of this has been already mentioned in article **171.**

The broach employed to finish off the holes should be gently rubbed with rouge in the direction of its length before use. This treatment removes slight blisters and renders its edges less cutting and less liable to leave small particles of steel adhering to the sides of the holes. They constitute a cause of wear which is but little noticed; it is, however, proved to exist by the fact that if, after continued use, the cutting edges of small broaches are examined by means of a powerful lens, saw-like indentations are at once detected.

After using the broach, the hole is polished by a fine point of peg-wood dipped in polishing rouge or in a paste of soft wood charcoal and oil. Some experienced watchmakers condemn the use of polishing rouge; we, however, as well as many others, have found it to work well, but it is essential that the subsequent cleansing be performed with care.

The pivot-hole in the cock which is traversed by the verge pivot must be of moderate dimensions, so that, in case a pivot breaks the train may be prevented from running down.

The nose of the potence and the counter-potence must be rounded in a tallow-drop form from the side of their end-plates, and there should be a slight interval between them in order that fresh oil may pass into the hole as that already there becomes dried up (**91**).

Jewelled Pivot-holes.

176.—Jewelled pivot-holes are not so satisfactory as those of good brass; they render a watch difficult to regulate and liable to sudden and unforeseen variations, and similar difficulties are incurred when the escapement pivots are made of extreme fineness.

As was the case with ruby pallets, the whole problem resolves itself into a question of friction; for timing depends on the existence of a certain equilibrium between the power and the resistance—hence, if the necessary conditions are secured with steel and brass in contact, they will no longer be maintained if the brass be replaced by a substance which is

much harder and better polished and, as a consequence, less *impressed* than this metal is under a given pressure.

177.—The reduction of the resistance due to friction renders the escapement very sensitive, and it becomes necessary that the whole mechanism be much more carefully proportioned. Watchmakers who are accomplished men of science may be successful in this; as to others, we can but express a hope that they will not engage in experiments which have objections greater than that of merely involving a waste of time.

The hole in the dovetail is easily thrown out by stopping and causes rapid wear to the pivot; a ruby hole would ensure the proper action of the escapement for a long period. The chances of its being beneficial will be gathered from the account subsequently given of some experiments made by a Paris watchmaker.

The Balance.

178.—Neither authors nor practical men have any definite geometrical rule for ascertaining the size and weight of a balance.

This difficult and novel question will be discussed at considerable length in the sequel; for the present we must merely give the reader the data arrived at by experience—data which, as will be seen, are at best exceedingly vague.

Its diameter in the modern form of watch should be, according to an empirical rule, about the same as that of the barrel. Experience, at any rate, indicates that a theoretically perfect balance will approximate to such a size.

As regards its weight, in watches as now constructed a balance which beats freely, without its balance-spring, less than 25 minutes an hour is too heavy and must be diminished; otherwise it would occasion setting, and a large amount of useless friction. One which beats more than 27 minutes is too light and therefore difficult to regulate; it must be replaced.

It must be observed that this operation, called in French *tirer les minutes*, in English *half-timing*, must not be attempted until we are assured of the soundness of the depths, the freedom of all the moving parts, the play of the pivots, and the presence of a proper supply of oil, etc.; for even a slight variation in the freedom of the train will influence the relation of the motive power to the regulator, and, as a consequence, the vibration of the balance, which, although beating the required

number of minutes, will thus probably prove to be incapable of ensuring an accurate timing.

Watches with a slow period of vibration require heavy balances and delicate balance-springs; hence they are liable to banking when carried. It is advisable therefore that the balance should not make less than 16,000 vibrations, or more than 18,000.

A balance must be in perfect equipoise; its weight concentrated at the circumference, and its arms and centre as far reduced as is consistent with solidity.

179.—If a balance beats a known number of minutes, say 20, we can calculate the extent to which its weight must be reduced in order that it may beat, say, 26 minutes. The following formula for solving such a problem is given by M. Henri Robert.

Square the number of minutes required (26) and the number already obtained (20), then calculate the proportion as follows: the first square is to the second as the initial weight of the balance is to its final weight *(x)*. The value thus obtained for *(x)* gives approximately the weight of the balance which will beat 26′ per hour (**1316**).

The Balance-Spring.

180.—It may be taken as a general rule, admitting of but few exceptions, that the uniform and perfect expansion of a balance-spring, that is, the regularity in the work performed by the several elements of its length, is the best test of its regulating power. In the majority of timekeepers in ordinary use, success in timing is far more due to this property than to its being of any particular length.

Without attempting here to discuss a question which will be treated of in considerable detail subsequently (see the chapter on *The Balance-spring*), we would simply observe that the verge watches of the present day, having much longer balance-springs than those of older construction, are yet in no way superior in the matter of timing. So far as this spring is concerned the reason of this shall be pointed out.

It must not consist of too great a number of turns: the opinion of the majority of practical men who have studied the subject, an opinion which is confirmed by our own personal experience, is that as a rule the verge escapement cannot be regulated well when provided with such a spring. Now the greater the number of turns composing a spiral of a given diameter, the greater must be its length, and it will require so

much the more motion to bring all its coils into action. Moreover, in ordinary watches we rarely meet with a long balance-spring absolutely homogeneous in all its parts, and capable of coiling and uncoiling with perfect uniformity. In this escapement the arcs of vibration are of moderate extent, and the internal coils as a rule do all the work, while the external ones are almost useless, and only help to render the positions of the working coils the more variable. Besides, during the longer arcs, or when a shake occurs in wear, the expansion of the spring is at times violently interrupted and the closely contiguous coils come in contact, especially if weak parts exist.

As the length of the spring is increased, the inconveniences above alluded to, as well as those occasioned by changes of temperature, become more appreciable. It is as well, therefore, to have about six or at most eight turns. This number should only be exceeded when it is impossible to find, within the limits here given, springs which expand uniformly.

The flat balance-springs of ordinary watches are not adapted for producing isochronal movement; and it will, moreover, be evident that isochronism would be useless in the verge escapement where the action of the spring is very constrained.

It would be an advantage if the spring could be slightly thicker towards the centre than in the outer coils, in order that with the slightest motion of the balance it might vibrate throughout its entire length; at the same time its impelling force must be the same throughout, since it is impossible that it should progress uniformly; for it will be obvious that if some coils are weak and some are strong, the former only will vibrate, the latter merely expanding to a very slight extent.

The diameter of the spring depends upon the radius of the balance. The stud is placed at a distance equal to half this radius from the centre of the cock.

The spring must be perfectly round and flat, and accurately centred on the verge. It should be firmly held both in the collet and stud. Finally, extreme care is required in the selection and attaching of a balance-spring; it is of the highest importance to the going of the watch.

To mark the Lifting Points.—To fix the Banking Pin.—The Drop.

181.—It has been already seen that when the angle of the pallets is 100° the entire lift should occupy 40°. Should it vary

sensibly from this amount, there will be too much drop if it is deficient, and in the converse case the drop will be too short and the balance will overbank. Hence the banking of the pallet against the face of the teeth will occur more frequently.

With too great a drop, motive power is wasted, the wear of the pallet is more rapid, and the timing is at fault (**46**).

Too short a drop renders the escapement liable to catch when the teeth of the balance wheel are at all worn, or when particles of matter adhere to the pallets or the points of the teeth. Besides which, there is much greater danger of the edge of the pallet catching against the back of the tooth.

182.—The following is the best method for MARKING THE LIFTING POINTS AND FIXING THE BANKING PIN.

When the balance wheel axis is accurately in a line with the body of the verge, a mark is made on the cock, exactly ove. the middle of the curb-pin slide; then, with a pinion gauge, one-third of the diameter of the balance is measured off and marked in front of the cock, where the two spots corresponding to the points of the gauge will indicate the extent of lift. The mark made in the first instance on the cock should be exactly midway between these two points. (The ratio of the diameter of a circle to its circumference being as 1 is to 3·1416, one-third of the diameter will correspond to rather less than 40° measured along the circumference.) Greater accuracy will be attained by employing the *grammaire* (see **507** and plate V, fig. 1).

The balance is now checked by a piece of paper, and the lifting action caused to take place on one side. Immediately on the tooth dropping, a mark is made with rouge on the edge of the balance immediately below the mark at the middle of the cock; the balance is then moved in the reverse direction, and from the second drop another mark is obtained. Midway between these is the point at which the banking pin should be fixed.

To Cut the Cock Bankings.

183.—To cut the bankings, the balance is so set that a tooth rests on the edge of the pallet and is just prevented from escaping. A mark is made on the rim of the balance opposite to that on the cock and, when the balance has been removed, the distance between the mark so made and the banking pin is determined by means of a pinion gauge; if one of its points be now set on the plate immediately below the central mark on

the cock, the other point will indicate the spot at which the cock banking should be cut on one side.

Proceed in precisely the same manner to determine it on the other side.

It would be useless to give, in addition to the above, the various methods employed by watch-jobbers in ascertaining the lift and banking, for an intelligent workman can always find out for himself the most expeditious means.

Effects consequent on Great Variations in the Motive Force.

184.—If the motive power of a verge watch be increased, the arc of oscillation is greater, and it gains; with a less force it loses, notwithstanding the fact that the arcs are reduced, because the balance is freer and moves less briskly.

Some watchmakers have attempted to apply this circumstance to procuring a kind of isochronism in the oscillations. They should, however, have remembered that the greater rapidity in the case of the long arcs is due to the banking of the pallet, and causes both wear and irregularity.

As regards the relation which, according to Berthoud, should exist between the recoil and lift in order to secure isochronism, it is useless to attempt its determination; for, besides there being great difficulty in doing so, it would require the wheel to fall just within the body of the verge; moreover, the wear of the holes would rapidly change the positions of the moving parts.

To examine an Escapement.

185.—It is impossible for a book to adequately replace the instruction given by a skilful practical man, and an intelligent workman can arrange a system of examination at once reliable and expeditious, but not until he has carefully studied the causes of stoppage.

In our little work entitled *The Watchmakers' Handbook*, he will find various tools described for ascertaining the opening angle, closing or opening the pallets, measuring their width, trying the balance wheel, adjusting the fusee, weighing the balance-spring, etc., etc.

We would here only point out that he must specially direct his attention to the exact division of the balance wheel and its perfect centring, the width and opening of the pallets, and

especial care is required in setting the two axes at right angles, as well as in securing perfect facility of recoil at the contrate wheel depth, etc.

CHAPTER IV.

CAUSES OF STOPPAGE AND VARIATION IN THE VERGE ESCAPEMENT.

The Wear of the Pallets.

The rapid wear which occurs at the surface of the pallets is one of the principal causes of the irregularities characterizing the verge escapement. As some watchmakers even now explain it on wrong principles, we will enumerate the circumstances generally accepted as giving rise to a destruction which is sufficient entirely to alter the action of the escapement.

Supposed causes of wear.

186.—Some authors attribute it to the fact that the pallets are struck successively by 11 or 13 teeth, and thus are subjected to more frequent impacts than these teeth themselves. They also argue that the perpetual percussion on one and the same point occasions a loss of polish and a pitting, while the point of the tooth wears without changing its form.

If such an explanation, which after all is not borne out on examination, were the right one, no verge could resist wear, but it is well known that they are occasionally met with in a remarkable state of preservation*.

One cause of wear, on which practical men appear to be agreed at the present day, is the presence in the brass of foreign particles introduced into the metal when molten, or to a film of particles of steel on the faces of the teeth derived from the cutter or file used in forming them; or lastly, to the minute imperceptible blisters concealing metallic oxides which, as is well known, always abound on the surface of the metal composing the wheel.

* When watchmakers are called upon to replace a verge of which the pallets are found to be in good condition, they are always extremely careful to simply clean the wheel without attempting any kind of polishing, etc.; for they rightly judge that, providing the pallets of the new verge are well made, they will, as a rule, resist wear for a considerable period. This is an additional proof that the wear of the pallets must be attributed to the bad quality of the brass, or to the careless construction of the wheel; it is also to a great extent dependent on the depth of the contrate wheel, for the simple stopping of its pivot-holes sometimes suffices to cause a verge to wear which had satisfactorily resisted all destructive action.

The presence of oil on the pallets, causing dirt to accumulate and the points of the teeth to become sticky, excessive drop, teeth cut too thin, or a wheel of too small diameter, the verge burnt or insufficiently hardened, oxidation of the wheel, bad workmanship (such for instance as sharp angles, the omission of the requisite polishing and burnishing, etc.), careless cleaning, and lastly, the employment of second-rate materials, are all causes which tend to increase the wear of the pallets.

187.—From experiments made under very favourable circumstances it appears that the existence of oil on the pallets is only a cause of wear in that it collects the dirt that finds its way into the watch and the small particles of metal which are removed from the points of the teeth by wear.

It has been observed that in verges cut short of the centre, the wear of the pallets is most rapid. But since the drop is in such a case less and the recoil greater than in others, this fact tends to prove that recoil is one of the chief causes of wear.

Had this fact been observed earlier, a bad contrate depth might, with reason, have been assumed to very frequently cause wear of the pallets, for it opposes a considerable resistance to recoil, and thus greatly increases the pressure on these pallets.

Formerly verges resisted wear for a longer period.

188.—Every watchmaker has observed how much more rapidly modern verges wear than those of older construction.

One author, referring to this subject, leaves the reader in doubt as to whether it is likely to arise from the superior quality of the metal then employed, the difference in dimensions, or the greater height of the cases, which allowed of the axes and verges being long and thin, so as to be slightly elastic. Probably by this means the friction does on the whole become less detrimental than when the axes are short and rigid.

The above passage, when published in our First Edition, attracted the attention of an intelligent Paris watchmaker, M. A. Ferrier, and led him to make the following curious experiment. He took a watch in which the verge pallets wore away with great rapidity, and so arranged the escapement that the balance-holes could yield slightly on applying a definite pressure to them. The watch has since been in action for two years without

the wear of the pallets taking place. It must of cause be understood that no regard was taken to its timekeeping properties.

Bearing in mind these facts the more rapid wear may be thus explained:

1. Formerly the balance wheels and verges were not made in a factory or in large numbers, and very great care was devoted to their construction; the wheels being formed of well-hammered brass from an old kettle, which during the long period it had been in use had experienced varying degrees of heat, and thus been refined by a number of operations of the nature of *pickling;* and the verges of the very best steel highly hardened and polished. This fact in itself would suffice to explain the superiority of the older productions over those of more modern date; for the system of piece-work being now adopted, the sole anxiety of the workman is to produce as much as possible. Thus it is that the wheels are nearly always formed of second-rate brass carelessly hammered, and for the same reason they are rarely divided with accuracy. As to the verges made on this system, the steel is often badly selected, and they are very frequently insufficiently hardened or burnt.

2. The greater thickness in old watches rendered it possible to employ a higher barrel, and to increase the diameter of the balance wheel.

As the impulse was thus communicated to the pallets by a lever of greater length, the friction occasioned less wear.

The mainspring was broad and thin, and so maintained its elasticity very constant, and one could therefore be selected of the exact strength required or nearly so. This cannot be done in the case of modern springs as they are necessarily made narrow and thick, so that they have but little flexibility, and soon lose part of what they have. They must then be stronger in proportion than formerly.

It should also be noted that with a thick watch, the verge may have a long axis and the oil will not ascend to the pallet.

3. The older watches beat, as a rule, a less number of vibrations than modern ones, and the arc of vibration was shorter (Berthoud recommends that it be three times the lifting arc). Latterly the number of oscillations and their extent have been increased, whereby the friction is magnified and the wear becomes more rapid.

4. The greater opening in modern watches causes the face

of the pallet to act in a plane more perpendicular to the axis of the wheel, as has been already explained, and the friction becoming greater occasions a more rapid wear of the pallets.

Means recommended for preventing wear.

189.—In the first place the depth of the contrate wheel should be such as to facilitate the recoil as much as possible.

The pallets should be very carefully polished and the precautions adopted which have been already enumerated in the article on the Verge.

As regards the balance wheel, no sharp angles should be allowed to exist, the faces of the teeth should be rounded and even polished, and the wheel then pickled in acid.

All the steel, as well as those parts of the wheel which it is desired to preserve from the action of the acid, must be coated with a thick layer of tallow, wax, or a soft paste formed of whiting and oil (care being taken not to place it accidentally on the points of the teeth). The wheel must be immersed for about ten seconds in nitric acid of the density 1·16 (that is, containing about equal parts of acid and water) and then washed successively in pure water and alcohol. It is well to complete the cleaning with a brush charged with rotten-stone or charcoal and oil, in the manner already described in the article on the balance wheel.

Some watchmakers plunge the wheel into the above mentioned acid several times, and withdraw it as soon as it has assumed a golden colour. Others place a drop of the acid on a sheet of glass, and invert the wheel in it, resting on the points of its teeth. The operation is completed as soon as the acid turns green, and the wheel must then be washed. A protecting coat should first be applied to the pivot.

Or one of the following processes may be resorted to; the pivot is protected by means of a small pellet of bread, and the points of the teeth are either held momentarily in the flame, or, after being first dipped in oil, they are made to rest on a blueing tray and heated until the oil volatilises, when the wheel is thrown into cold water.

Lastly, the practice of slightly touching the points with a piece of pure wax, just before putting the watch together, is said to have a beneficial effect in preventing wear.

It is frequently sufficient to carefully clean the surfaces in the manner explained in a previous paragraph (**172**).

When a cause of wear exists which cannot be mitigated by any of the above precautions, any watchmaker can, from the details already given, easily judge for himself as to the best course to pursue in the case under consideration.

Setting of the Balance.

190.—A setting is generally due:

1. To a want of freedom in the train, caused either by bad depths, the thickening of oil, weakness of spring, unnecessary friction, etc.

2. To a balance which is too heavy, or checked in its motion by the pivot-hole being too small, the friction of the verge axis in the potence-hole, or against the edge of the dovetail, or else to bent pivots, etc.

3. To a balance-spring which is badly centred, or out of flat, and so prevents or constrains the motion of the balance.

4. To the pallets and teeth being too deeply pitched.

5. To a balance wheel having too much endshake, and consequently at times falling against the body of the verge.

The causes which occasion stoppage in any particular instance must be carefully ascertained before any correction is applied, and the unjustifiable practice of diminishing the weight of the balance until the watch does not set should never be resorted to. Those who make such a mistake seem to be unaware that the balance, becoming too light, will be very greatly influenced by variations of temperature, the shakes which necessarily occur in wear and the inequalities in the motive force; that the watch can never be properly regulated, and that they are only adding one more fault to those which had already occasioned the stoppage.

Other causes of Stoppage and of Irregularities in Timing.

191.—A BAD FOURTH OR CONTRATE WHEEL DEPTH. This depth must be adjusted with the greatest possible care, as we shall subsequently explain. (See the Treatise on *Depths*, article **1166**.) It alone causes at least half of the stoppages and irregularities which occur in verge watches.

WANT OF ADJUSTMENT BETWEEN THE FUSEE AND MAINSPRING. In the absence of a proper tool for adjusting the fusee, the only possible remedy is to change the spring until one is found corresponding approximately with the fusee throughout its length.

A VERGE CUT BEYOND THE CENTRE. It must be replaced (**170**).

INCORRECT OPENING OF THE PALLETS. The angle must be increased or diminished by means of a lamp, but without changing the temper of the pallets, by one of the methods known to every watch-jobber. (They are given, and the necessary tools described in the *Watchmakers' Handbook.*)

A VERGE WITH TOO NARROW PALLETS. This fault, which is characterized by an excessive drop, may be neutralized by slightly increasing the verge opening, and causing the teeth to fall nearly up to the body of the verge. A safer plan would be to replace this latter, and so avoid all risk of the opening being too great.

CATCHING OF THE TEETH ON THE EDGE OF THE PALLETS, especially if these are filed too sharply. This fault occurs very frequently when the verge opening is considerable. The balance, steadied by a small piece of paper, should be turned round until the banking pin rests against the cock; then by moving the wheel backwards and forwards it is possible to ascertain whether there is sufficient play between the points of the teeth and the verge.

THE VERGE BODY NOT CONCENTRIC WITH ITS PIVOTS. The wheel under these circumstances will engage more with one pallet than with the other, although from micrometer readings their width may appear to be identical.

A VERGE WHICH IS NOT CYLINDRICAL AND TOUCHES EITHER THE SIDE OF THE POTENCE OR PIVOT OF THE BALANCE WHEEL.

A BALANCE WHEEL WHICH IS NEITHER TRUE NOR ROUND. Such a fault detracts from the lift and causes the drops to be unequal. The wheel must be made true tooth by tooth, but the best way is to replace it.

BADLY FORMED OR IRREGULAR TEETH. The effect is the same as in the last mentioned case.

It is sometimes possible to partially nullify the above sources of irregularity by the following artifice. The balance wheel is set sufficiently near to cause it to catch slightly. The teeth which pass with difficulty are marked with rouge, and are subsequently bent in the required direction by a pair of strong polished tweezers. This operation requires considerable care and must be continued until the requisite lift is obtained.

TEETH WHICH ARE TOO MUCH OR TOO LITTLE INCLINED. Teeth when not sufficiently inclined should be reduced on the face with a file which must not be allowed to touch the points. When the inclination is excessive, the fault cannot be satisfactorily

remedied, since it is impossible to remove enough from the back to avoid the catching of the pallets. It may be worth while attempting to bend the points slightly backwards.

A WHEEL WHICH TOUCHES THE BALANCE COLLET, OR ELSE ENGAGES TOO NEAR THE BASE OF THE LOWER PALLET.

OVERBANKING is due to a badly placed banking pin, or to the corners of the cock being cut too deeply. When the balance wheel teeth are irregular or badly formed, overbanking may occur only with certain teeth.

CATCHING occurs when the drop is too short, the teeth irregular, or their points too thick.

THE TRAIN NOT FREE. It sometimes happens that the wheels, especially the balance wheel, are found to be locked when the watch is put together. This circumstance is due to cockling of the top plate which, through being bent, does not rest evenly on the four pillars.

PIN STRIKING AGAINST THE BANKINGS. When the edges of the cock are not sufficiently cut away, the banking pin in its wrong place, or the lift or motive force excessive, this fault may be met with.

THE BALANCE WHEEL AXIS NOT PARALLEL, OR THE VERGE AXIS NOT PERPENDICULAR TO THE TOP-PLATE. This will cause a loss of lift, and the teeth will overlap one pallet more than the other.

A BALANCE WHICH IS NOT POISED, OR OF INCORRECT WEIGHT, TOUCHES OR RUBS ON THE COCK OR AGAINST THE STUD, ETC.

AN INFERIOR BALANCE-SPRING, OR ONE BADLY FIXED, OR WHICH PRESSES AGAINST ONE OF THE CURB PINS, ETC.

A REGULATOR SLIDE WORKING TOO EASILY, OR GETTING OUT OF PLACE.

A COLLET OR STUD THAT IS LOOSE.

A BANKING PIN WHICH COMES INTO CONTACT WITH SOME PART, OR CATCHES AGAINST THE COCK.

PIVOTS TOO SHORT. When their extremities do not work against the endstones, so that the conical shoulders rub against the sinks.

Lastly, to all these causes of stoppage or variation, must be added BAD DEPTHS; such are frequently met with, and they render the force impelling the escapement excessively variable. From Sully's experiments it appears that an increase of one half in the motive force produced, in the watches of his day, a difference of six hours in every twenty-four (**184**).

CYLINDER ESCAPEMENT.

CHAPTER I.

Preliminary.

192.—The Cylinder Escapement, one of the class of frictional dead-beat escapements, was invented by the famous English watchmaker, Graham, about the year 1720. It came into notice in France in 1724, and was at that time considered of but little value by some French watchmakers. This arose from a distrust of anything novel, and from the fact that most of the earlier experiments were unsatisfactory, owing to its principles not having been at that time accurately laid down. F. Berthoud, among others, attempted, but in vain, to demonstrate from theoretical considerations the superiority of the verge over the cylinder escapement and, in general, over all dead-beat escapements. He was opposed by Jodin, who in a small work containing much valuable matter, likewise of a theoretical nature, gave evidence of remarkable insight and a most careful study of this escapement, but who, although faultless in the greater part of his argument, did not throw much light on the details of construction of an appliance which was still only in its infancy.

At the present day, after being the subject of such a series of ill-directed experiments as follow on every important discovery, the cylinder escapement is better understood, and it is more perfectly constructed in consequence of the introduction of special tools, which both facilitate its manufacture and increase its accuracy; for the purposes of every-day life it now gives excellent results.

Berthoud was the first to clearly lay down rules for the construction of this escapement; but his theory of dead-beat escapements in watches is generally admitted at the present day to be erroneous in many particulars. Nevertheless, credit is due to him for his work, and in reading it we must remember that at his time, owing to an opinion accepted without question amongst watchmakers, the escape-wheels of cylinder watches were formed of brass of considerable thickness, the oil was rapidly decomposed, and the teeth, in the case of light wheels, were liable to be strained by the touch of the workman, and wore away with far greater facility than does a steel wheel under the same pressure. The cylinder was usually thick and

heavy, and the balance light but of considerable diameter. The pivots, as a rule thicker for a given sized cylinder than they are at the present day, worked in brass holes which gradually became larger and so changed the relative positions of the working parts. It will be evident that under these circumstances the cylinder watches of that period were sure to be characterized by excessive and very variable friction, and the irregularities, especially those due to changes of temperature, were much greater than occur in modern watches.

If we observe that as a rule the arc of vibration of the balance was shorter than that now in vogue, that the number of vibrations formerly never exceeded 14,000 to 16,000 per hour, and at the present day is increased with advantage to about 18,000, and that the use of the fusee, then almost universal, had the triple objection of being useless, of involving a needless increase in the motive force, and of causing, more especially when the oil was at all thick, a setting at the time of winding, it will be evident why this escapement gave rise to so great a diversity of opinion among watchmakers of the last century.

Such a diversity of opinion is no longer justifiable at the present day, when we can study it confidently and with profit by the light of long and well directed practical experience, and with the science of machine-construction in a highly advanced condition.

Advantages of the Cylinder Escapement.

193.—As compared with the verge, the cylinder escapement offers the distinct advantages which are common to the majority of dead-beat escapements.

The fourth wheel of the train being flat, ensures a much safer depth than in the case of a crown wheel, that fruitful cause of variation.

The suppression of the fusee reduces the liability to disarrangement and, since less force is required in the prime mover, renders it possible to use a weaker mainspring, which will therefore be thinner and more elastic.

Lastly, if sufficient care be devoted to its construction, this escapement will, notwithstanding the absence of the fusee, render a watch a far better timekeeper than can be secured with a verge movement. And this evidently shows that the cylinder escapement counteracts, with sufficient accuracy, irregularities occasioned by the mainspring in the force transmitted through

the train. This correction, arising as it does from physical and mechanical causes to be considered later on, is mainly secured by properly proportioning the several parts of the escapement to each other.

Its cost is moderate, its construction is not very difficult, and it can be applied to thin watches without diminishing its accuracy.

When the escape-wheel and cylinder are formed of the best steel, thoroughly hardened and polished, the escapement will work well for a long period without wear of the rubbing surfaces. It will only require cleaning and the application of fresh oil from time to time, and, except in case of accident, will maintain its timekeeping properties. It is quite a mistake, however, to class it, as some ignorant watchmakers do, with the escapements capable of very accurate timing. For even at its best, although excellent in ordinary watches, it is inferior to the duplex, lever, and chronometer escapements, where scientific accuracy is required.

Inconvenience of the Cylinder Escapement.

194.—The main objection to this escapement consists in the fact that rather frequent cleaning is necessary, because oil cannot be dispensed with and the friction during rest occurs at such a distance from the axis of rotation (on the internal and external surfaces of the cylinder); the thickening of the oil increases the friction which is already considerable. From this it follows that the motive force transmitted through the train becomes weaker as the resistance opposed to the escapement is increased, and the oscillations, being impeded, will gradually be made shorter and slower, and the watch will have a losing rate.

This slight inconvenience, and the need of more frequent cleaning, are to a great extent compensated for by the superior timing of a cylinder over a verge watch.

Denomination of the several parts of the Cylinder Escapement.

PLATE I. FIGURE 1.

The Cylinder.

195.—1. Axis or arbor.
2. Great or top plug.
3. Small or foot plug.
4. Great Shell.

5. Small Shell.
6. Plug Face.
7. Great or Engaging Lip or Edge.
8. Small or Disengaging Lip or Edge.
9. Small opening or banking slot.
10. Flat of the shell.
11. Cylinder rest or half-shell.
12. Cylinder column.

The collet which carries the balance is attached to the great shell, as shown at *s v* (fig. 3, plate I.). The same figure gives a side view of the cylinder with the plugs inserted, and ready to receive the balance on *s* and the balance-spring collet on *v*.

The Escape Wheel (fig. 15).

196.—1. Flat of the tooth.
2. Inclined plane or simply incline or impulse curve.
3. Pillar.
4. U-arms.
5. Point of the tooth.
6. Heel of the tooth.
7. U-space.

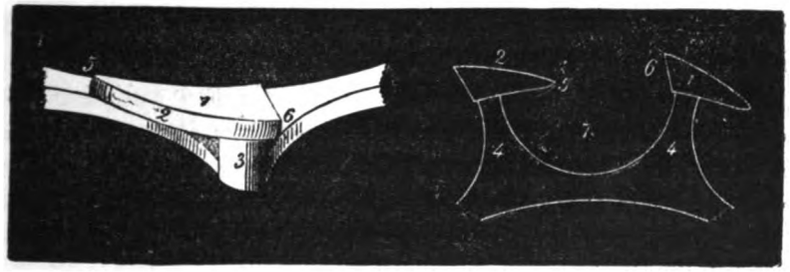

Fig. 15.

ACTION OF THE ESCAPEMENT.

197.—Assume the mainspring to be let down and the cylinder to be held at rest in one of the U-spaces by the balance-spring.

Now give a turn to the watch key: a tooth of the escape-wheel comes in contact with the great lip of the cylinder *a* (fig. 2, plate I.). The tooth continues to move in contact with the lip, and, pushing it backwards through the entire length of the

impulse curve, compels it, with the balance attached, to turn towards the right. This sliding of the impulse curve on the lip constitutes a *lift*. At its conclusion the tooth is suddenly released from the lip with a slight jump (the *drop*), and its point falls against the internal face of the cylinder (*b*), which continues its rotatory movement through the momentum imparted to the balance. During this period the *first rest* occurs, for the wheel remains stationary in consequence of the point of its tooth being held in contact with the interior of the moving cylinder (see *b* and *d*).

This state of rest will be maintained until the cylinder has been brought to its original position by the action of the balance-spring. It will then, however, cease, because the inclined plane, meeting with the second lip (*f*), will slide past and push it backwards (*g*), so producing the second *lift;* the tooth then finally leaves the cylinder, as seen at *k*. As soon as one tooth escapes the next (*h*) comes in contact with the external surface of the cylinder, and occasions the *last rest*, which continues until the cylinder is brought back by the action of the balance-spring. The tooth is thus in the position (*a*), and acts in precisely the same manner as we have seen the first act; the same is of course the case with all succeeding teeth.

It is thus evident that each tooth gives rise to two rests, one on the outside and one on the inside of the cylinder; and two lifts, one on each lip. The sum of these lifts is the total lifting angle of the escapement.

Each impulse curve therefore impels the cylinder alternately to the right and left, thus both setting it in motion and restoring to it at each impulse the force necessary to maintain its vibrations.

The greater the pressure exerted by the wheel the more vigorously will it act on each lip, and any change in the motive power will cause an increase or diminution in the extent of the vibration, but is entirely without influence on the extent of lift.

The slot known as the *banking slot* (9, fig. 1, plate I.) prevents the lip from striking against the U-arms and enables the balance to perform a complete rotation.

CHAPTER II.

PROPORTIONS IN VOGUE AT DIFFERENT EPOCHS.

198.—JODIN * (1754—1766). "The point of the tooth should, in its motion, pass through the centre of the cylinder, and the friction during rest will then be less detrimental."

The incline should be straight, or its curvature should approximate as nearly as possible to a straight line. "*The movement imparted to the balance will in this case be a maximum.*"

Lift 40°. It might with advantage be reduced to 30°.

Cylinder aperture, about 190° measured at the half-shell.

Thickness of the shell about $\frac{1}{8}$th of the length of the inclined plane.

It should be here observed that, in his day, cylinder watches were usually large and thick, and in consequence of the employment of brass escape-wheels it was beneficial to have a shell of considerable thickness, so that the friction with the incline occurred at more than one point.

Jodin was the first to point out that success in the timing of horizontal watches depends on the correct proportioning of all their parts; more especially in the ratio between the escapement itself and the weight and size of the balance. Doubtless from a regard to his own private interests, he did not publish the results of his experiments on this subject.

He pointed out the bad effect which an excessive force of impact on the surfaces of rest has on the timing; whether that result is due to the incline being too short or badly formed.

He also was the first to show that "When in a watch of this construction, the balance is too large, the watch goes slower with an increase of the motive force; conversely, when it is too small, it goes faster."

199.—The question as to the best form to be given to the teeth of the escape-wheel is far from being a novel one, as some

* Jean Jodin was a well-known French watchmaker of the last century. If his work has not attained the position to which the valuable matter it contains entitles it, this is due to the fact that Jodin frequently connects very sound and important observations with entirely false reasoning; his style is affected and diffuse; and the frequent errors and the misapplication of terms weary the reader of this little volume.

recent writers would imply, for it is a century since Jodin discussed it and, we must add, discussed it with intelligence, although not exhaustively. His work contains some errors which, however, are due rather to the state of horology and of science at his time, than to the ignorance of the author himself.

The following are extracts from his work:—

"The three definite forms which can be given to this curve (the incline) correspond to the three following, the lift in each case being assumed to be the same:

"1. The form of the curve may be that already discussed (that is, a curve such that equal portions of its length cause the cylinder to rotate through equal arcs) so that a uniformity in the arcs occasioned by the several portions will result.

"2. The curve may be of such a nature that the same resistance is offered by the balance-spring throughout; for it is evident that any spring offers more resistance when wound up than it does when released.

"This latter basis of adjustment is a very attractive one, and the curve is capable of considerable development" (as has been seen often enough since).

"3. The curve may be such as to cause the balance to perform the greatest possible arcs of oscillation, in which case it most nearly corresponds to a straight incline.

"Lastly. A fourth form of curve may be contemplated, such that the advantages above pointed out might co-exist and secure at each point of the curve an approximately perfect regularity of movement, but we are obliged to forego it," etc.

"I can imagine I see the reader embarrassed in his selection; he is afraid lest he take a false step; the ends secured are equally attractive. There is a curve of which each portion gives an equal arc, another which compensates for the want of uniformity in the action of the balance-spring, and a third which gives the greatest possible arc of oscillation to the balance.

"For a long time I myself was undecided between the first two advantages, but ultimately the second prevailed. A curve possessing such hopeful properties (always being subjected to the same resistance on the part of the balance-spring) could not but commend itself to a lover of logical sqeuence, and it was only after very careful study that I succeeded in making some of the very worst of cylinder escapements; for this was the one curve of all the three that was to be avoided.

"I went back to the first, whose object is to secure equal arcs for equal portions of itself; but not being any better satisfied with it, I gave it up and directed my attention to that which logically appeared to be the most hopeless; I mean the approximation to a straight plane, where the impulses given by the force at successive instants seemed to be so very irregular.

"As regards the first it is a matter of no importance that each portion of the incline gives rise to an equal arc of oscillation. This is not a case in which we are concerned with the equal subdivision of the circle; *our object is to ascertain the best means of communicating motion and, at the same time, of communicating the greatest possible amount of motion* (except, as will be shown in the sequel, when the objections to such a course counteract the advantage which would result from an increase of a few degrees in the extent of the arc of oscillation).

"In considering the advantages of the second form of curve, is it not natural to assume that the object is to nullify a considerable resistance by a very slow movement, and that it is of importance, for the attainment of that end, to save all the impelling force, not gaining it at one point only to lose an equivalent amount at another?"

200.—LEPAUTE (1755—1767). "It would perhaps be well to reduce the amount of lift in the cylinder escapement; say, for example, to 15° on either side, for observation shows that the more we reduce the lifting arc in a dead-beat escapement, the greater the complete arc of oscillation becomes: now the greatness of the arcs is just as important an advantage in watches as their smallness is in clocks." (This is to a certain extent true, but not entirely so.)

It is unnecessary to give the rules laid down by Lepaute at length for, like the above, they are merely borrowed from Jodin.

201.—P. LE ROY (1761). "The precautions taken by the best watchmakers in the construction of the cylinder escapement are as follows. 1. The wheel must be light, the points of the teeth must pass through the axis of the balance, and the inclined planes must be such as will give a lifting arc of from 30° to 40° (the exact amount of this angle will vary with the motive force employed, and other circumstances). 2. On escaping from the edges of the cylinder, they must drop against its internal or external face, but this contact must last for as

brief a period as possible. 3. The drop should be as small as practicable," etc. 4. The balance must not be too small, for then it becomes sluggish and increases the variations caused by a change of position.

"Judging by experience, I consider that the number of vibrations should vary from 16,200 to 17,000. The latter number was usually employed by Graham in his watches."

202.—F. BERTHOUD (1763—1786). "The middle of the incline (*b*, fig. 16) should pass through the centre of the cylinder, so that this plane may, in producing the lift, act to the same extent before and beyond the line of centres *(a b)*. The resolution of the forces will thus be, very approximately, the same on either side of this line, and uniformity will be secured. *

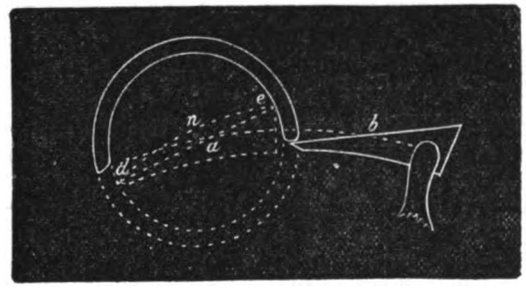

Fig. 16

"When the middle of the incline passes accurately through the centre of the cylinder, the drop is of less extent and so less force is wasted.

"The half-shell should be equal to a semi-circumference of the cylinder, plus the height of the inclined plane. Thus, the more the teeth are inclined, the more the cylinder will be closed, and the lift will, of course, increase in like proportion. It becomes necessary, therefore, at the outset to decide what amount of lift should occur on each lip, and, the number of degrees being once known, it is only necessary to add them to half the circumference of the cylinder."

As Berthoud considered 20° of lift to be necessary on each lip, the half-shell of the cylinder should, according to

* Even when this equality is not accurately secured no harm will result, as we see from the detent escapement in which the lift only occurs on one side, and the single hook escapement, which, from its long vibrations necessarily involves a greater amount of lift on one side than the other; an equal division is, however, preferable. (*Note by Moinet*).

him, be half the circumference (360° divided by 2), plus 20°, that is 180° + 20°, or 200°.

He made the thickness of the shell equal to $\frac{1}{10}$th the length of the inclined plane.

"The angle between the inclined plane (*d a e*, fig. 16), and the tangent to the arc *a b* at the point *a*, should not exceed the number of degrees (20) added on to the semi-circumference, as explained above; or, if the tangent be drawn at *d*, on the curve *b a* continued, the angle must be only half the required lift, since the line *d e* is twice *a e*, the radius of the cylinder.

"Thus, for a lift of 20° on each lip, the angle between the incline and the tangent at *d* must only be 10°."

Berthoud recommended pivots of some thickness, so as to reduce the extent of the vibration and the friction on the cylinder, but expeirence has proved this to be a mistake.

We will limit our quotations to what is given above, for the other recommendations and theoretical opinions originated by this author have for the most part been shown by experience to be fallacious.

Notwithstanding all that experience taught him, Berthoud adhered to his mistaken views, doubtless impelled by an unfortunate conceitedness; to the reader of his *Supplément à l'Essai sur l'horlogerie* this will be evident, for he characterizes the cylinder escapement as *very bad*, and conceives it to be more faulty than the verge.

Towards the end of his life he made some excellent cylinder watches. Those into whose possession they have fallen will see that they refute the opinions of their author as well as the great majority of the rules of construction he laid down in his writings, which extended over a period of more than thirty years. It would have been better for his reputation in the future had he honestly confessed himself to have been in error.

203.—F. CALLET* (a paper presented to the Society of Arts of Geneva, and published in 1780). Callet considers that if we desire the movement of the cylinder, when acted on by the escape-wheel, to be the counterpart of that of this wheel, the incline must be curved, in part concave and in part convex, when acting on the entrance-lip; and, when acting

* A skilful calculator, born in 1744 at Versailles, and died in 1798. He constructed some tables of logarithms.

on the exit-lip, must be slightly concave. But this inclined plane evidently cannot be changed while passing from one lip to the other, and the author, therefore, concludes: "Perhaps watchmakers would do best if they adopted a straight incline."

The curvature should be varied when, for a given diameter, the number of teeth is changed. In the case of a wheel with fewer teeth the first curve is more pronounced, whereas the second approximates to a straight line.

Two successive lifts cannot, therefore, be precisely similar in this escapement, and the two resulting vibrations must be of unequal extent.

This author observes that the form of the impulse curve should be such as will communicate a considerable movement to the balance. For if the impulse be slight, and the pressure during rest excessive, the effect of the balance-spring may be nullified. "Hence, of all the inclinations that might be given to the impulse curve, *there exists one that produces the greatest possible oscillation of the balance*, etc. When the angle is very acute, the lifting arc becomes too small, and the lip of the cylinder may very easily be so carried out of reach of the teeth that the impulse curves do not act on it at all. It becomes important, therefore, that their inclination be such that every point of the curve may give an impulse to the cylinder-lip, but this must not be carried to an extreme. The exact inclination can only be determined by experiment."

204.—FÉTIL the elder* (1802). "Several dead-beat escapements have fallen into disuse before they have been properly examined and explained, so that we can give no definite information about them; we must be content, therefore, with a few observations. Thus, for example, little has been said as to the *cylinder* escapement, at one time so much in vogue, further than to discredit it; attention has not even been drawn to the following main points for consideration:—1. The point of contact during rest, which must be as nearly as possible tangential; it therefore follows that the number of the teeth of the escape-wheel is not a matter of indifference. 2. The diameter of the cylinder, as compared with that of the balance, and indeed with the movement as a whole; and this is important,

* Pierre Fétil was born at Nantes, about the year 1753. He died at Orleans, on the 18th of May, 1814.

not so much from the point of view of the working of the escapement, which is chiefly influenced by the first condition above referred to, *but in order that its several properties may be so inter-related as to secure regularity in the going of the machine to which it is applied.*

Fétil adds that a watchmaker of his day pretended to avoid wear in his cylinders by giving the impulse curves of the teeth of the escape-wheel (in brass) an "irregularly convex surface, such that the curvature was greater at the middle than at the two ends of the curve, thus causing the latter portion of the lift to be produced by a curve of but slight inclination: the cylinders were cut away to the extent of about one half, so that, after the lifting action had taken place, the point of each tooth would just fall on to the resting surface; and from this it followed that when the cylinder had taken up the position midway between two successive lifts, that is, the position it would occupy if only under the influence of the balance-spring, a quarter or even a third of the impulse curve would be found to have passed the cylinder edge; this form only gave a lift of 40°, and sometimes even less."

While endeavouring to find some satisfactory explanation of this method of construction, Fétil foresaw that the choice of the metal must, in some way, influence the final result.

205.—JURGENSEN (1805). "The cylinder must be cut away 20° short of the diameter." Thus the half-shell will measure 200°.

Lift 40°. "The middle of the impulse curve should pass through the centre of the cylinder.

"In order to avoid setting (in fusee watches), the inclines of the teeth of the escape-wheel are curved, the curvature being considerable towards the point and gradually shading off towards the heel of the tooth (whereby the resistance occasioned by the balance-spring is overcome with greater ease). At the same time this curve must approximate to a straight line, for experience has shown that *the greatest attainable movement of the balance is secured when the incline is straight.* In watches arranged to go during winding up it is needless taking precautions to avoid setting; the inclines may as well, therefore, be straight.

"The cylinder must be formed of steel of the best quality, such as that used for fine gravers. It is well to make it of such

a thickness that the lips can be accurately rounded without any tendency to forming cutting edges, as that would cause a pitting of the inclines, which in turn would wear the cylinder itself." (According to the plate that accompanies the work, the thickness of the shell would be about $\frac{1}{10}$th of the length of the impulse curve; this is hardly in accordance with the text.)

The interval between the two plugs should be sufficient to ensure the oil against any undue attraction on their part, for if such took place it would leave the wheel and occasion a rapid wear of the escapement.

Jurgensen set the escape-wheel in such a manner that the plane of the teeth was not at right angles to the axis, and thus the teeth came in contact with the lips at different heights. This arrangement was first suggested by Romilly.

Prior to Jurgensen's time the escape-wheel was always constructed of brass, owing to an erroneous idea which was generally accepted without question. He was the first to anticipate and to demonstrate experimentally the advantages possessed by a steel wheel.

206.—TAVAN*. (In a report submitted to the Society of Arts of Geneva, in 1805, but not published till 1830, after the text had been revised and corrected.)

"The rubbing faces or the impulse curves of the teeth should be inclined at an angle of about 24° to the tangent to the circle described by their base. Such an inclination appears to ensure that the lift takes place under the best conditions possible.

"They should have a slightly convex form, the radius of curvature being identical with that of the wheel itself. Such a form is found to be most advantageous in practice.

"If we compare the lift produced by a straight incline and one curved as above described, we see that, in the first case, *two-thirds* of the length of the tooth will be expended to produce 10° of lift, and the remaining 10° will result from the *final third*. But it is just during this period that the balance-spring opposes the greatest amount of resistance (this statement is perfectly true if a balance-spring starting from a condition of rest is under consideration, but ceases to be so when the spring is under the influence of a motion already acquired). In the

* Antoine Tavan was born at Aost, in France, in 1749. He went to Geneva at the age of twenty, and died there in 1836, after attaining considerable reputation as a watchmaker.

second case with a circular curvature the action is much more uniform.

"The incline, whether straight or curved, should be rounded off in the direction of its length, and thus possess a *beaded* form. The extent of surface that rubs against the lips of the cylinder will thus be very much reduced, but experience has shown that this does not occasion any increased wear, for the wear which does take place is generally due to dust attaching itself to the oil on the teeth and forming a scratching mass. It often also results from the metals employed being of an inferior quality or badly polished.

"The thickness of the shell should be about one-seventh the length of the inclined plane. As regards the cylinder opening, the portion left (or half-shell) should exceed a half-circumference by 10° on either side, plus the amount by which the rounding of the corners of the lips occasions a loss of leverage during the lifting action."

Half the circumference (180°), together with twice 10°, gives 200°. Adding a little on account of the rounding of the lips, we conclude that the half-shell is to be slightly over 200°.

"The lift should be 20° on either lip, making a total of 40°.

"From the very nature of the movements that take place between the rubbing surfaces of this escapement at one time, the frictions at the two lips of the cylinder are not similar; in the first lift, the lip is already moving in a direction somewhat opposed to the advancing impulse curve of the tooth, whereas, in the second, the two mobiles are travelling in the same direction with different velocities." (The first kind of friction is that now called engaging, and the second is disengaging friction.)

"As the friction during the rests takes place alternately on two radii of resistance differing from each other by the thickness of the cylindrical shell, it will be evident that the influence they have on the vibrations of the balance must be unequal."

207.—MOINET (1846). "As a matter of fact it is usually the practice to make the half-shell or resting part of the cylinder, including in this term the rounded edge of the entrance-lip and the incline of the exit-lip, 190° at most, following the recommendation of Cumming*; so that immediately on the heel of the tooth escaping from the entrance-lip, having im-

* An English watchmaker who, according to Moinet, was the first to propose (1776) that the inclines should be curved. He also advocated the amount of

pelled it backwards, its point drops on to the resting surface, but only at such a distance from the exit-lip as is necessary to ensure a rest taking place, that is to say, so that it shall not drop direct against the incline of the lip; for this purpose about 3° of circumference should be allowed.

"If to this 3° be added the slight lift occasioned by the incline of the exit-lip or the rounded edge of the entrance-lip, which is about 5°, it becomes necessary to add 8° to the semi-circumference, and it follows from this that 188° is the exact extent of the half-shell.

"As it is difficult to be very accurate when working on such a minute object, this may be taken to be 190°, which should be slightly reduced if experiment prove it to be necessary.

"It is evident that the drop of the point of the tooth on the outside surface of the shell should take place with the same certainty as on the inside, that is to say, about 3° from the rounded edge of the lip; this curvature should extend a little farther on the external than it does on the internal face, and should also be less rapid on this outer surface where the tooth is on the point of entering the cylinder."

As regards the amount of lift and the inclination of the impulse curves, Moinet recommends the same proportions as are given by Berthoud.

After referring to "another modification introduced in the cylinder escapement and revived at different periods," namely, the adoption of a curved in place of a straight incline, Moinet observes:—

"As regards the choice of form for the incline, whether straight or curved, and the complicated inter-relationship between the power of the balance-spring and the velocity of the balance, the safest practice would be, as we have already pointed out, to determine experimentally which form procures the greatest supplementary vibration; for, in questions of this nature, *argument may very easily mislead, and it is therefore safer to resort to experiment.*"

He reproduces the proportions adopted in a watch by Berthoud, and observes that the diameter of the cylinder is

opening above given, and considered that the radius of curvature of the incline of the exit-lip should be half that of the impulse curves themselves.

We would observe that Jodin discussed the question of curved faces several years before Cumming, so that on this point the French watchmaker has the advantage of priority.

about $\frac{1}{20}$th of that of the balance, adding: "others make it $\frac{1}{14}$th, so as to increase the compensation caused by frictional rests."

208.—M. WAGNER (1847). "The most important improvement that can be introduced in any escapement is to cause it to work, under given conditions, with the least possible amount of friction." (The question of friction, although important, is not of the first importance, especially if only its indefinite diminution is principally considered. See **106** and following articles.)

According to this author, in order that a cylinder escapement may not set, that the friction may be reduced to a minimum, and that the extent of the oscillations may be as great as possible, it is essential that: (1) The centre of rotation of the cylinder be on the tangent;* (2) the cylinder be cut away 180°, minus the amount by which the lips project (about 4° or 5°); (3) the lifting angle be equal to the circumference of the wheel divided by twice the number of its teeth.

With a 15-tooth wheel, then, the lifting angle will be 12°.

"The surface of the tooth (or the straight line passing through its two extremities), the diameter of the cylinder, and the tangent will therefore be in a line.

"I would add that when the above conditions are maintained, this angle (the lifting angle) varies inversely with the number of teeth of the wheel.

"The adoption of these proportions will in all probability be opposed by many practical men who, with a wheel of this number of teeth (15) have been in the habit of giving 20, 25, 30 and even 40 degrees of lift, under the impression that by so doing a greater impulse is communicated to the balance; it is easy, from the law of the inclined plane, to demonstrate the converse to be the case." (Experience is opposed to the opinion of this author, as it proves the impulse to be increased in the great majority of cases. The law of the inclined plane is not the only one bearing on the subject.)

"As the height of the incline is increased, the space traversed by the balance becomes greater and, as a consequence, the friction during the lift; if the cylinder be enlarged the friction during the supplementary arcs will also increase."

When the height of the inclined plane is thus augmented,

* That is, a tangent drawn to the path of a point of a tooth where it cuts the external surface of the cylinder.

"if it be desired to maintain the opening of the cylinder at 180°, its centre of rotation must be in the middle of the incline of the tooth (assumed straight)."

But, "in this case, since the above condition of the centre being on the tangent is not carried out, there will be a still further and quite appreciable increase in the friction.

"It is nevertheless possible to retain the centre of rotation on the tangent, and at the same time to vary the lift from the amount above mentioned; but this can only be accomplished at the expense of the cylinder opening."

As regards the form of the impulse curve, the author recommends a convex surface, *a b c*, considerably divergent from a straight one, and formed of two arcs of circles united, and having their centres at U and T (fig. 17).

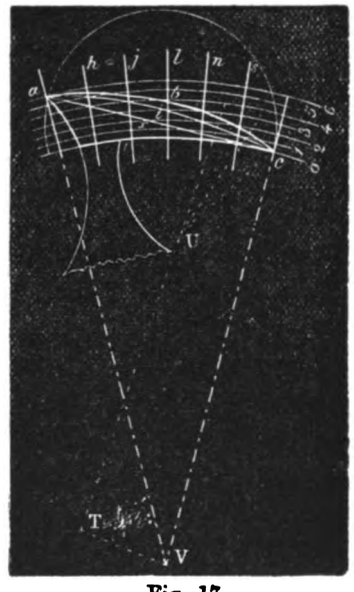

Fig. 17.

This curve has, according to M. Wagner, the property of rendering the action of the motive force proportional to the increasing resistance of the balance-spring.

We would observe that Jodin (**199**), after making a number of experiments on curves of this class, abandoned them in favour of a straight or very slightly convex incline.

209.—M. HENRI ROBERT (1849). The amount of lift which M. Robert considered sufficient for the cylinder escapement does not exceed 25° or 30° "at most."

He concluded that, as regards the form of the incline,

a straight one or a concave curve *approximating very nearly to it*, imparts the greatest arc of vibration to the balance.

"The more marked the curvature is at the commencement of the lift, the greater will be the force required to give motion to the balance; afterwards the tooth will slide against the edge, only producing a slight amount of lift, and the drop will be the more violent according as the curvature is more marked at first, and the displacement of the cylinder towards the end is less. Thus, the excessive force necessary to impel the wheel at the commencement of the lifting action increases the liability of the lip to wear, and causes the point of the half-shell on which the drop occurs to become pitted sooner."

According to M. Robert, "every curve will, during the lifting action, bring a greater number of points successively in contact with the cylinder than a straight line does, since its length is greater than that of the chord subtending it, and consequently the clogging of the oil will be all the more appreciable."

Table of the Proportions recommended by different Authors.

OPENING OF THE CYLINDER.

Angular measurement of the half-shell:

Jodin—Lepaute	190°	Minimum, 185° Maximum, a little over 200°
Berthoud—Jurgensen	200°	
Tavan	a little over 200°	
Moinet	190°	
M. Wagner	185°	

TOTAL LIFT, INDICATING THE HEIGHT OF THE INCLINED PLANE.

Jodin—Lepaute	30°	Minimum, 24° Maximum, 40°
Berthoud—Jurgensen	40°	
Tavan	40°	
Moinet	40°	
M. Wagner	24°	
M. Henri Robert	25° to 30°	

Various Observations and Summary of the Chapter.

210.—We do not intend to give a critical examination and comparison of the data contained in the above table, as the reader can easily do this for himself after reading the following articles. We will only here recapitulate the present chapter in a few words.

The amount of opening recommended by Jodin towards the middle of last century, is precisely that advocated by Moinet nearly a hundred years afterwards. Some watchmakers of Jodin's day opened the cylinders rather more than he did; the half-shell

was, therefore, very nearly the same as that proposed by M. Wagner.

Jodin, who in the first instance adopted 40° as the lifting angle, subsequently reduced it to 30° in accordance with an opinion then current that it was advisable to diminish the lift as much as possible in the case of a dead-beat escapement (**200**); we shall subsequently see that as regards the cylinder escapement of his day, he was justified in limiting it to this amount. In our own time, M. Robert has suggested this latter figure of Jodin's as a maximum limit for the extent of the lift.

The earlier makers caused the point of the tooth to pass through the axis of the cylinder "so that the resting action might take place perpendicular to this axis, and the friction therefore be more uniform." They evidently, even at that period, had recognized the advantage of tangential rest.

211.—As regards the form to be given to the impulse curve, they had already, before the commencement of the present century, proposed or investigated: (1) the straight incline; (2) a slight curve towards the point of the tooth; (3) a curve most pronounced at its centre; (4) considerable curvature at the point of the tooth; (5) a sort of winding or concave curve; (6) two arcs of circles united, one described with half the radius of the wheel and the other with the entire radius.

Ever since the invention of this escapement, each author who has discussed it with any attempt at completeness, except Jodin and Fétil, has laid too much stress on some one special feature of the problem, giving it an extravagant importance without apparently perceiving that success depended on the proper harmonizing of the whole; and further, that the desired end might be attained in several ways, each, however, requiring that not only one, but all the conditions of the problem should be satisfied.

From what precedes it will be evident that the rules which should guide the watchmaker in constructing the cylinder escapement were foreseen by Jodin, and their salient points distinctly indicated by Fétil. How then are we to explain the fact that, after an interval of sixty years, several authors have wasted time by doing over again the work of last century, when a few experiments properly conducted would have sufficed to demonstrate the worthlessness of solutions that are merely geometrical (**77**)?

CHAPTER III.

PRINCIPLE OF THE CYLINDER ESCAPEMENT.

212.—The classification of escapements as recoil, dead-beat, and detached escapements is, as has been already pointed out, imperfect; we shall, however, retain it since it is usual, and, although not strictly logical, we shall employ the expression "escapements *with frictional rest*" when speaking of those in which the wheel, after communicating an impulse to the balance, remains at rest, being held by a moving body that forms a part of the axis of the balance. Such, for instance, are the cylinder and duplex escapements in watches, and the pin, anchor and Graham escapements in clocks.

The misuse of the term "scientific principles."

213.—The cylinder escapement can be regulated under conditions *apparently* so variable that it has given rise to considerable discussion, and has been the occasion of a great amount of writing both in pamphlets and treatises. Many of these, penned by ill-educated practical men, are characterized by such senseless expressions as "theory requires so-and-so, but practice says no;" "Men of science know nothing about horology, and yet they *officiously dictate laws for its guidance*," etc., etc.; and one soon perceives, on taking up the works of these authors, who for want of real authority put forward their ignorance as a justification of their attempting to teach, that they are only trying to lisp a language that is unknown to them, and are speaking of subjects they do not properly understand.

If a practical man publishes details of the most approved methods of manipulation, and points out the proportions that have given him the greatest satisfaction, explaining everything in such detail as to be clearly understood, he renders a distinct service to his art, and proves himself to possess real merit; but if he is not well-informed in the exact sciences, he must avoid discussing them, and not risk a plunge into theoretical subjects, since any mistake would endanger the reputation of his work. Another objection to the books we refer to is that they mislead young students by classing as *scientific principles* what are

nothing more than empirical rules applicable only in certain special cases.

We have intentionally italicised the above expression, which is a favourite one with numerous authors. In virtue of these scientific principles, which none of them define, one demands one thing, and another another!

Hippocrates says yes, but Galen says no.

Then after a train of argument, more or less specious, that goes in a circle round the main issue, and seldom really copes with it, we come back to the indisputable phrase: "If the mechanism be constructed *on scientific principles*, we obtain," etc.

These are indeed remarkably elastic principles: they adapt themselves to every system, are applied to the most dissimilar contrivances, and do not conform to those logical and unyielding laws which must regulate the movements of all bodies. These false principles would have practical men believe some antagonism to exist between theory and practice, between the head and the hand.

It is needless for us to refute such an absurdity; men of intelligence will see it in its true light, but to the younger watchmakers we would say: "This antagonism does not exist; *science never says one thing and practice the opposite.*" We must not expect of theory more than it is able to give, and the contradiction which, according to certain books, exists between it and practice, results solely from the author not having studied his subject with sufficient care and completeness, so that he charges theory with decisions for which it is not responsible.

An excellent tool may be clumsily handled; this is an exactly parallel case to the one we are considering, and we have spoken in no spirit of criticism, but simply because the errors to which the practice referred to have given rise, hinder the progress of horology, and prevent it from receiving, at the hands of truly learned men, that consideration which it deserves.

NEW THEORY OF ESCAPEMENTS WITH FRICTIONAL REST.

214.—The following theory depends upon no gratuitous hypothesis; it is based upon the logical application of laws that have been clearly proved, and on numerous carefully observed phenomena.

It is borne out in all its detail by the experience of the most skilful watchmakers, and explains the origin of those intermin-

able controversies that have occupied the attention of artists for many years past.

It is the result of uninterrupted labour during a period of ten years. We do not mention this fact for the unprofitable pleasure of satisfying vanity, but so as to show that, however anxious and determined one may be in the pursuit of truth, it is difficult to keep clear of the hazy theories so generally received; theories which are only supported by one-sided arguments, and which in the great majority of cases are disproved in practice.

The same laws must govern the Escapements of Clocks and Watches.

215.—The construction of escapements with frictional rest, whether they be for clocks, timepieces or watches, is governed by the same physical and mechanical laws. We must, however, always remember this important difference, that in the case of the pendulum, the regulating action depends on a force which never varies in a given locality, namely gravitation; whereas when an annular balance is employed, we are compelled to rely on the elasticity of the balance-spring; and this is a force that varies considerably with changes of temperature, and can be caused to alter at will in a manner very analogous to that represented by the law of vibration of the pendulum.

The problems in the two cases therefore become comparable, the only difference being that a certain *unknown* is determined for the pendulum by means of the law of gravitation, and for the annular balance by the law of elasticity.

Functions of an Escapement with Frictional Rest.

216.—The functions which every escapement is required to perform are:

1. To moderate, and at the same time regulate the velocity of rotation of the train of mechanism.

2. To restore to the moderator the small amount of force that it has lost at the conclusion of each complete oscillation.

3. To effect this restitution of force in such a manner that all the oscillations occupy exactly the same period of time. The attainment of such a result does not in any way preclude the possibility of employing unequal movements as regards the actual space traversed.

Its Action is Composite.

217.—The action of a frictional-rest escapement may be shown to consist of two effects; the strain or impulse on the

impulse pallet or edge, and the pressure on the axis of the moderator while in motion; an action and a reaction which will be found discussed in the chapter of the Introduction, headed: *The Lift* of Escapements (**94**).

It is of the utmost importance that each of these two effects be separately unravelled, in order that we may ascertain in what cases the two actions unite, and in what cases they wholly or in part neutralize each other.

Propositions summarizing the new theory.

With a view to abridge our work, and to enable the reader to understand it at a glance, we will at once lay down the bases of the theory.

They are summed up as propositions, which severally point out a principle to be theoretically and practically proved in the sequel.

FIRST PROPOSITION.

218.—If the motive force be exactly counterbalanced with an escapement arm, the conditions of equilibrium will be maintained, however the length of this arm be varied, providing the lifting angle and motive force remain unchanged.

SECOND PROPOSITION.

219.—If the lifting angle and the motive force remain constant, the force of the impulse that maintains the movement of the regulator will increase with any diminution in the length of the escapement arms.

THIRD PROPOSITION.

220.—The resistance occasioned by friction on the resting surfaces is proportional to the lengths of the radii of rest

FOURTH PROPOSITION.

221.—The lift and the motive force remaining the same, the period of an oscillation will change with any variation in the length of the escapement arms.

FIFTH PROPOSITION.

222.—There is one, and only one, length of the escapement arms that is adapted for ensuring the nearest possible approximation to isochronism in the oscillations.

SIXTH PROPOSITION.

223.—The useful effect of an impulse, measured by the amplitude of the arc of vibration described by the moderator,

varies with the height of the incline by which this impulse is communicated.

SEVENTH PROPOSITION.

224.—There is one degree of inclination of this incline which secures, with a given motive force, the greatest extent of oscillation with the greatest regularity.

EIGHTH PROPOSITION.

225.—If the relation between the impulse lever and radius of rest that is best suited to any given pendulum or annular balance has been determined, it will be necessary to vary their proportion with any change in the dimensions of this moderator.

NINTH PROPOSITION.

226.—The size of the escape-wheel is not a matter of indifference. It is directly dependent on the weight and velocity of movement of the wheel, on the effective height of the impulse curve, and the friction that occurs on its surface.

227.—A full demonstration of these propositions is impossible except by the aid of mathematics of an advanced description. But without pretending to a strict proof, which after all is not absolutely necessary here, we will proceed to demonstrate the truth of the above laws by means of the elementary mechanical principles given in the Introduction. We shall explain how they can be confirmed by experiment, and skilled workmen will be able to multiply the evidence thus afforded by constructing the apparatus referred to with all the care and accuracy that they have learnt to devote to instruments of precision.

As a prelude, and in order to enable every watchmaker to benefit by the explanations that follow, which, as will be seen, are within reach of anyone of average intelligence, we will explain a graphical method applicable to the study of the resolution of forces in an escapement. We shall follow the system proposed by M. Rozé (junior) at a meeting of the Paris Horological Society.

Resolution of Forces in Escapements.

Pressures.

228.—The object of every escapement is to change the nature of a movement.

The character of this transformation depends upon the mechanical details of construction of the escapement.

Now, as the impacts and friction that occur in every piece

of mechanism in action occasion a loss of motive power and, consequently, a variability in the amount of force transmitted, it is important to examine the relations that exist between the arrangement adopted in the escapement, and the amount of force necessary to neutralize the exact quantity of friction which that arrangement involves.

By observing the manner in which the forces are resolved we arrive at this information (**67**). Let O and A (figure 1, plate III.) be the centres of movement of the wheel and the escapement arm (the pallets, radius of cylinder, etc.), K H is a tangent to the lifting face, either rectilinear or curved, at the point M whose action is under consideration. At this point draw M U perpendicular to H K. Assume the motive power to be applied at the point M of a tooth of the wheel, and represent it by M P drawn perpendicularly at the extremity of the radius M O. We will consider the arm as fixed in the position indicated in the diagram.

In consequence of the action of the tooth on the plane K H the motive power will tend to force apart the points O and M, and the pressure exerted on them can be ascertained by drawing the parallelogram S M U P, using M P as a diagonal, and the radius of the wheel and M U the normal to the lifting surface as two contiguous sides.

This construction shows the manner in which the force is resolved into two portions; one of these, M U, acts perpendicularly to the lifting surface opposed to the tooth, displacing it, and the other, M S, is directed against the pivots of the wheel and converted into friction.

The force M U is thus the only portion that produces a useful effect; that is, fitted to induce a motion of the escapement arm when free to move.

An examination of figure 1 (plate III.) clearly shows that this force, M U, will itself be resolved into two parts as soon as the motion of the lever commences: one, M T, acts on the pivots of the pallets or cylinder, that is on the fixed point A, and the other, M Q, produces a motion of rotation of the lever round this fixed point.

The above explanation is equally applicable whether the lift takes place entirely at the end of a lever or in part on the tooth itself; these are the cases of a lever and a horizontal escapement.

By drawing the escapement in the position it occupies at

different periods of the lifting action in the manner above explained, we shall be able to study the resolution of the force during the entire lift, and to ascertain for each successive point the relative values of the pressures exerted.

The length of path.

229.—Besides taking account of the relative intensities of the forces, it is necessary also to observe the relative distances traversed by the several working parts, for any change in the velocities will occasion a change in the manner in which the motive force is distributed.

Figure 2 (plate III.) gives a method of obtaining a simple expression, sufficient for our purpose, for the amount of sliding action that takes place during the lift.

Let O and A be the centres of movement of the wheel and escapement-arm, connected by the line O A. Let M be the point of application of the force at the moment under consideration; in this case it represents the extremity of a tooth of the wheel. Let H K be the tangent at the point M, so that it coincides with the impulse curve at this point. At M draw M G perpendicular to H K, and cutting the line of centres, prolonged if necessary, at G.

M. Rozé has demonstrated by calculation that, with a uniform lifting action communicated at M, the angular velocities are in the inverse ratio of the lines O G and A G, or the distances of the point G on the perpendicular M G from the centres of movement, measured along the line of centres.

We can at once make an application of this theorem:—

If the ratio of the angular velocities remains the same for all the points of lift, the normals (perpendiculars) drawn at the points of contact that correspond to different positions of the lever will all pass through one and the same point situated on the line of centres or its prolongation. In figure 2 (plate III.) G is the point thus fixed upon.

THEORETICAL & EXPERIMENTAL PROOF OF THE ABOVE PROPOSITIONS.

If a given motive force be maintained in equilibrium by the arm of an escapement it will remain at rest, however the length of this arm be varied.

230.—The lifting angle of the triple escapement-arm A *b c* D (fig. 3, plate III.) is in each case D A *l*, so that the three inclined planes *h g*, *n f*, *m l*, which, when the escape-wheel and the amount of lift remain unchanged, correspond to three arms

of lengths 1, 2, and 3 respectively, will be contained between the lines D A, A *l*.

The three projections, *h s*, *n t*, *m* D, are of equal thickness, since they depend on the distance apart of the teeth of one and the same wheel; hence the inclination of the impulse planes must vary with the length of the lever, diminishing as it increases.

"An inspection of figure 3 (plate III.)," says M. Wagner in his Memoir, "shows that for a given arc of oscillation the incline *h g* is three times as rapid as the incline *m l* which is at thrice the distance from the centre of oscillation, because the heights *h s*, *m* D, remain the same while the base D *l* is three times as long as *s g*.

"It follows that if, through the greater inclination of *h g*, the projection at distance 1 is displaced with three times the force exerted on *m l*, the force repelling the latter projection will be thrice as effective as that repelling *h g*, because the lever arm A *o'* is three times A *o*, and thus the force lost by reducing the inclination is gained by the increased length of the lever arm; and conversely, the force gained by the greater inclination is neutralized by reducing the length of the lever arm."

231.—He concludes that, ignoring the question of the friction on the resting surfaces, the length of the escapement arm is unimportant "*since, however it be varied, the force of the impulse remains the same.*"

And, if we take into consideration this friction on the resting surfaces, "*the length of the escapement-arms must be reduced as much as possible,*" since the impulse remains constant while the friction on these surfaces increases with the length of the arm.

232.—We cannot admit this double conclusion to be legitimate for, from our point of view, it does not in the least follow from the given premises; it is quite a mistake to assume that to be a general law which is only true when the escapement is at rest and the resistances simply in equilibrium with the motive force.

An escapement in equilibrium and at rest does nothing, and might as well not exist for any useful purpose that it serves; the condition of the problem is entirely altered when the system is set in motion. From being statical it becomes dynamical.

We shall have to revert at some length to this important subject, but before doing so will again quote from M. Wagner (nephew), who was, to the best of our knowledge, the first to

discuss and solve the theorem that forms the heading of this article; a theorem which is only one particular case of the general law of the equilibrium of forces.

FIRST EXPERIMENT.

233.—"Theoretical conclusions must always be supported by experimental results; for this reason I have constructed an instrument consisting of an escapement with arms of different lengths, by means of which the practical truth of the above principle can be demonstrated. The same apparatus enables us, moreover, to measure the friction with each length of arm" (he is only speaking here of the friction on the resting surfaces).

"The instrument (represented in figure 4, plate III.), consists of a double frame *b b*, mounted on a base *d d*, and supporting at *a* the axis of an escapement arm which is provided with very fine pivots in order to diminish friction; three projections, *i*, *j*, *k*, of equal height, are formed on this arm, so that the same wheel can act on them, and their distances from the centre of oscillation are as 1, 8, and 16. The three projections have the same amount of lift, a fact that may be verified by means of a divided arc of a circle attached below the longer projection and an index fixed to the base *d d*. To the axis of the escapement is fixed a horizontal arm *t*, at the extremity of which can be suspended a small balance-pan, the attachment being such as is employed in a delicate balance. Lastly, on this same axis is another vertical arm carrying an adjustable weight *r* by which the apparatus can be exactly brought into a state of equilibrium; when, however, the apparatus is thus adjusted, the balance-pan is disconnected from the arm *t*, being supported by its knife-edge on a piece fixed to the frame *b b*; the arm *t* only comes into contact with it at the moment when the arm *c c* of the escapement commences to move backwards, through the force exerted by the finger F (representing a tooth of the escape-wheel) on one of the inclined planes; this tooth F moves on an axis *o*, forming part of a sliding piece P, which can be fixed at any given point along the upright *v*, so as to bring F opposite either one of the projections of the escapement arm. Marks on the upright *v* facilitate the placing of this tooth accurately at such a height that it shall act tangentially to each projection. The stem of F is traced over with a thread to carry a mass *m* by which the action of the tooth on the inclined planes can be increased or diminished.

"Finally, in order to reduce as much as possible the causes of error, I constructed all the parts with the greatest possible care. The pivots were of extreme fineness, the rubbing surfaces very highly polished, and the apparatus could be accurately adjusted as to position by means of levelling screws *x x*.

"The following is the method of experimenting:

"After having set the tooth F to act on one of the projections, the weight necessary to maintain the apparatus in equilibrium when it thus acts on the inclined plane, is placed in the pan of the balance; then the tooth is presented to another projection, and the apparatus still remains in equilibrium with the same weight in the pan. Yet the shorter of the three arms is only one-sixteenth of the length of the longer; we therefore assume that intermediate arms, however numerous, would behave in precisely the same manner.

"The experimental result then confirms the theoretical principle that, ignoring friction, the length of the escapement arms is a matter of indifference."

234.—This conclusion of M. Wagner is in accord with the laws of statics, and quite true from a simply geometrical point of view, but it becomes erroneous when applied practically; for we must repeat what we have already said (**232**) that an escapement must always be considered when in action, and then the conditions of the problem are very different from those above assumed.

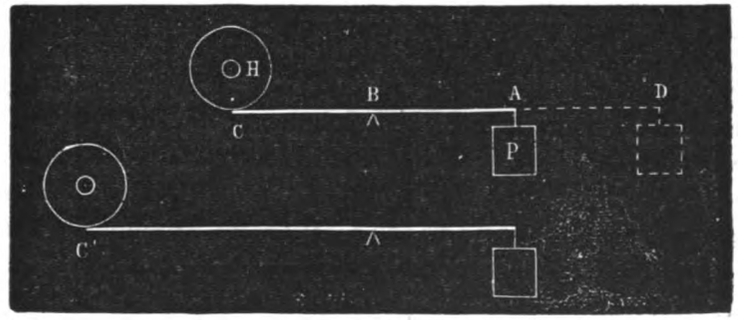

Fig. 18.

By concluding from the foregoing experiments that the escapement arm should be *as short as possible*, their true import has been overlooked, as we shall proceed to show; it is first necessary, however, to prove the following proposition.

The Resistance caused by Friction on the surfaces of rest is proportional to the Radii of Rest.

235.—From the law of the proportionality of friction to

pressure (**132**), it follows that every friction may be represented by a weight.

Let there be a lever C B A (fig. 18), movable on a centre B, and of the two equal arms B A, B C, let one B C rest against a rotating cylindrical body, and the other B A carry a weight P.

This weight may be taken to express, or to be a measure of, the resistance which the friction of the lever opposes to the rotation of the cylinder H. The intensity of this friction will be doubled by transferring the point of application of P from A to D, since B D is twice B A, or the same effect will be produced by applying twice the weight at A.

Conversely, if the radius of friction B C be doubled and the cylindrical body be transferred to C^1, the friction on the point of contact will be reduced to one half.

SECOND EXPERIMENT.

236.—M. Wagner was the first to practically ascertain, by means of the apparatus described in article **233**, and shown in figure 4 (plate III.) the amount of friction on the resting surfaces.

"If the tooth F be placed on any point of one of the curves of rest, a certain weight in the balance-pan will be required to overcome the friction thus occasioned, and to set the escapement in motion. If now this same tooth be caused to rest on a surface having a greater radius, we shall observe that a greater weight is necessary in order to neutralize the friction, and in the converse case a less weight will be required; it is thus seen that this weight increases directly with the length of the escapement arm, and will, in a corresponding proportion, diminish the freedom of movement."

The impelling force required to maintain the acquired movement of the balance, increases as the arms of the escapement are shortened.

237.—As in the preceding cases, let the angle of lift and the motive force remain the same.

Assume for the present that the impulses are really equal. It necessarily follows that, as the arms of the escapement are lengthened, the total pressure on the resting surfaces of the moderator will gradually increase while the impulse remains the same; whence it results that the motion of the moderator will be more and more impeded, and its oscillations will be slower and slower. The watch or clock will therefore lose more and more in its rate as we increase the length of these arms

If, then, this theory be true, the retarding effect to which we allude will become greater as the length is increased from *zero* upwards. Double the amount of motive force will occasion an impulse nearly twice as great, but since the pressure on the resting surfaces will be two, three, and four times as much when the arms become twice, thrice, or four times the length, the resistance should increase in the same ratio when the impelling force remains constant.

In practice, however, the converse of this is found to be the case, and it is easy to prove that it should be so by a simple application of the laws of mechanics.

238.—The advocates of very short lifting arms have either ignored or overlooked two elements in the problem under consideration: at present we will only discuss one of these. Although it is not easy to ascertain its exact amount in watches with escapement arms of the dimensions ordinarily met with at the present day, it can be determined without much difficulty when the dimensions are considerable.

The element to which we refer is the amount of force that is absorbed in friction of the tooth on the inclined plane, and in friction at the pivots.

An examination of the lines v, v', v'', v''', in fig. 3 (plate III.), is sufficient to show that the friction on the incline acts at a constantly increasing angle as the lever arm is made longer, from which it necessarily follows that the pressure on the arms and on the pivots becomes greater, and the amount of friction progresses in a like manner (**70** and **132**).

The principle of the resolution of forces (**228**) must be resorted to in order to solve this question.

Let r' and r (fig. 3, plate III.) be the centres of one and the same wheel, acting successively on the arms A h, A m of an escapement; $o\ r'$ and $o'\ r'''$ represent the radius of this wheel, and v', v''', the perpendiculars at the middle of each lifting surface. In the line A o drawn at right angles to the extremities of the radius of the wheel, let $o\ u$ and $o'\ u'$ indicate the amount of motive force exerted by this wheel, and draw, in the manner already explained, the parallelograms $o\ i\ u\ p$, $o'\ i'\ u'\ p'$.

The same motive force is in the first case resolved in the proportion of $p\ o$ to $o\ i$, and in the second case as $p'\ o'$ is to $o'\ i'$.

Now $o\ i$ and $o'\ i'$ represent the pressures exerted by the wheel at o and o'.

Approximately, therefore, from the figure, the pressure impelling the lever when the wheel is centred at r' is about 9, and less than 11 when it is at r'''.

The advantage gained by using a long arm is thus seen to be considerable in this first period of the action.

The force exerted at o can be resolved into two parts proportional to the lines $o\ u$ and $o\ q$.

One of these represents the pressure on the pivot A, and the other $o\ q$ may be taken for purposes of comparison to give a measure of the actual impulse communicated to the arm.

The parallelogram $o'\ u'\ p'\ q'$ shows in like manner the resolution of the force at o'.

If these several lines be compared, remembering that the friction is proportional rather to the adhesion (**42**) than to the pressure, and that the surface over which the friction occurs is three times as great in the case of the longer lever; if it be further remembered that the latter always weighs from three to four times as much as the shorter arm in order to secure an equivalent amount of rigidity, we shall conclude that the effective impelling force is about 4 with the arm A m and 5 with A h. The latter is therefore the most efficient.

We would again point out that the above values are only approximate; they are given thus in round numbers to assist in the explanation which it is unnecessary, however, to extend, since the reader will be able to do so for himself should he think fit. It is sufficient for our purpose to have proved that the impelling force derived from a constant motive power *increases*, disregarding friction during the rest, *in a definite proportion as we shorten the escapement arms.*

239.—The demonstration of this fact justifies us in laying down the following new law, which is sufficient to finally settle the prolonged discussion that has taken place with regard to the best length for the escapement arms.

The law may be thus enunciated:

The impelling force increases as the length of the escapement arms is diminished.

Now we know that the resistance due to pressure on the resting surfaces varies directly with this length.

Hence from these two laws, the following important conclusion may be drawn:—

Since there are two effects which vary in contrary direc-

tions with a gradual change in the length of the arm, there is necessarily one point at which they *most nearly approach each other;* that is to say, a point at which the disproportion between the two actions (the impulse and the resistance) is a minimum.

The several functions thus inter-related are not only very remarkable, but their consequences are so important that one is astonished at attention not having been earlier directed to them.

240.—We have now to discuss the most delicate portion of the whole question under consideration; the measurements involved are infinitely minute and very difficult to detect except by the slight variations which they occasion in the going of the watch.

What has been proved above is accurately true (providing the nature of the metal, the condition of its surface, etc., remain the same) only on the supposition that the axis of the lever is a mathematical line, that is to say, has no thickness and is incapable of displacement.

Such a condition of course could not exist in practice, and between the mathematical axis of suspension, which is at the centre of the pivot, and the point of contact of this pivot, there will be a distance equal to half its diameter.

This point, moreover, is constantly being displaced.

Consider figure 19, where all the dimensions have been purposely exaggerated.

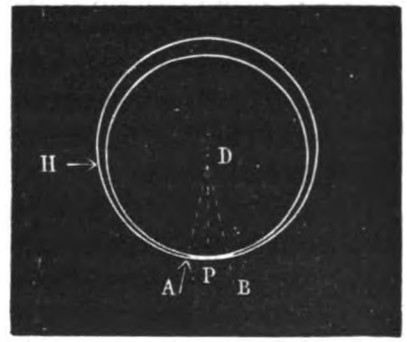

Fig. 19.

Let D be one of the pivots of a pair of pallets seen in the pivot-hole that supports it.

When everything is at rest the pivot will touch the pivot-hole at P. This point will be the fulcrum about which the resistance of the lever acts.

Assume the pallets to be very long. When the wheel begins to act on the left-hand side, the point of contact will be slightly displaced and, with a lift of average extent,

will be transferred to about A. The pressures during both the locking and the impulse act in nearly the same vertical line when the arms are long, and thus the point of contact will be approximately the same during both periods.

But such is no longer the case if the arms are very short.

The pressure on the impulse plane then acts laterally, and it follows that the arm, being held by the pendulum crutch, is compelled to transfer its fulcrum from A towards H, and occasionally even beyond that point.

The effect of this will be easily seen. The locking surface which is concentric with the pivot will for the instant become excentric, and while in this condition it must cause an acceleration in the movement such as is observed in recoil escapements.

This change in the point of contact, whether it be due to the shortening of the arm or to an alteration in the crutch, is deserving of careful study.

When the pallets are attached to the axis of the pendulum, it is the point of flexure of the suspending spring that varies.

241.—It follows immediately from the law given in article **239**, and the considerations contained in the last article that:

Any increase in the motive force will cause a watch or clock to *gain*, the arc of oscillation remaining the same (and even sometimes if it is increased), whenever the arms of the escapement are *too short;* and, conversely, to *lose*, if these arms are *too long*.

And this directly leads us to conclude that:

Since a single cause, namely *increased motive power*, can produce two opposite effects, there must exist some point at which these two effects neutralize each other, or very nearly so: and that, in taking this point to be the limit of the length of the escapement arm, the insensibility of the escapement itself to variations in the motive power is rendered as great as possible.

Theory is thus able to give a definite answer to the question we are discussing. It only remains to examine it practically, to subject the new laws to experimental verification, and the problem may then be considered to be finally settled.

THIRD EXPERIMENT.

242.—In order to confirm by experiment these theoretical deductions I have made several series of observations. I shall subsequently refer to them in detail, and at present merely describe the three following:—

In the first, my object was to show that if force is restored to a moderator which it has previously lost under different

conditions as regards pressure during locking, a gain in the rate occurs when this pressure is least.

I replaced the escape-wheel pinion in a cylinder watch by a pinion provided with a long and uniform shoulder; on this shoulder either of two escape-wheels could be accurately fixed, being firmly held by friction.

Their diameters were equal, but they differed considerably in the height of the impulse curves.

That least inclined closely approximated to the tangential position. Its friction on the resting surface was therefore much less than in the case of the other wheel.

The centres of the inclines were in the same positions in the two cases, and the action of the watch with these wheels was as follows.

The mainspring was wound up only *one* complete turn:

The watch beat 17,000 vibrations:

With the incline tangential, and a mean entire vibration of 190°,

In 55 minutes 45 seconds;

And with the more sloping impulse curves and a mean oscillation of 220°,

In 55 minutes 34 seconds.

It should be observed that the motive force remained the same, and the gain occurred while the balance was performing its longer vibrations.

The mainspring was now wound up *four* complete turns.

The same watch beat 17,000 vibrations;

With the incline tangential and a mean vibration of 220°,

In 55 minutes 42 seconds;

With the greater inclination and a mean vibration of 245°,

In 55 minutes 28 seconds.

In the second case as in the first, the longer vibrations were performed more rapidly than the shorter ones.

243.—These preliminary results I consider to prove with sufficient accuracy, that a change in the slope of the impulse curve causes a change in the relative intensities of the pressure on the resting surfaces and the impulse given to the escapement arm; and that, when the force remains constant, the differences must result solely from a more or less energetic lifting action, since the pressure on the resting surface cannot alter.

The conclusions at which we have already arrived might

be contested if they were only based on such observations as the above, since we ought to eliminate the complicated action of the balance-spring, which varies with the extent of the arc described by the balance.

We will therefore only take account of the two experiments in which the amplitude of the arcs was the same. The action of the balance-spring may be considered to be the same in both cases, and so may be eliminated.

When wound up one turn the spring neutralized a weight of 40 grammes.

When wound up four turns the spring neutralized a weight of 60 grammes.

Thus the same amplitude of vibration, 220°, was obtained,

With the incline set tangentially, by a force proportional to 3;

With the greater inclination, by a force proportional to 2.

It is hardly necessary to observe that the fact of the arcs being of equal extent in the two cases proves clearly that the force restored to the moderator is the same. But the movements necessarily possess more activity when the pressure (during the locking) is least in proportion to the impelling force.

If it be remembered, moreover, that, in the experiment in which the inclination of the impulse curves is least, the escapement acts tangentially, the fact of the increased rate being due to a change in the proportion between the impulse and the pressure will be placed beyond doubt; and, for the present, this is the only conclusion we desire to draw.

FOURTH EXPERIMENT.

244.—We were carrying on these researches and experiments when a fortunate coincidence occurred which confirmed our theory and gave us the support of M. L. Vérité, of Beauvais, a mechanician whose authority on this subject is unquestioned.

This ingenious artist had for some time past observed that the results arrived at in practice do not always confirm the predictions or rather calculations of the geometrical theory.

In order to clear up his doubts he constructed an apparatus, the description of which we defer until we have had an opportunity of inspecting it. It is formed of two identical clock movements; one of these can be moved vertically so as to cause the escape-wheel to successively act on pallets placed at different distances from the centre of suspension of the pendulum. The

two movements are brought into exact accord before this position is altered, and it will be evident that if a change in the length of the lever arms, on which the escape-wheel acts, causes variations in the rate, these will immediately be rendered manifest by the clock hands.

We will only add that by means of this happy combination, M. Vérité has established the fact that by increasing the motive force, the extent of oscillation remaining the same, the clock *gains* on its rate with very short escapement arms, and *loses* when these arms are too long.

His well-deserved reputation as an experimenter, is a sufficient guarantee of the value of these determinations and the truth of the conclusions to which he was led. They are in complete accord with our own results as well as those recorded in the following article.

FIFTH EXPERIMENT.

245.—This form of experiment is due to M. Kessels, of Altona, an artist whose work is fully sufficient to justify the European celebrity to which he has attained.

The inference to be drawn from the observation, first described in 1848, does not seem to have been grasped by any author.

Kessels established the fact that pallets with long arms, as were formerly employed, "have the grave defect of interfering with the isochronism, since the wheel causes a losing rate by its action on the locking surface," and this can be verified by simply disengaging the pendulum from the escapement when it is immediately observed to gain on its rate.

After having very appreciably shortened the pallet arms (from 30 lines to $5\frac{1}{2}$ lines), he showed that he had nearly attained to isochronism, and he found "a clear proof that this escapement, when working nearer to the centre, does not, in the smallest degree, interfere with the natural oscillations of the pendulum," which, when left independent of the movement, had a slight tendency to *lose* amounting to a fraction of a second in twenty-four hours.

The effect would have been more marked with shorter arms, but the above result, obtained frequently by Kessels, is quite sufficient, since it has been proved by the observations of so practised an horologist, with very perfect means of verification at his disposal, that when the pallets are long there would be a

loss and when very short a *gain* in the rate of the attached pendulum as compared with that of the detached one.

Kessels seems to have been hardly aware of the discovery which he was on the verge of making, for, after admitting that the impulse communicated to the long and short arms was the same, he observes: "The main point is that the action on the locking surfaces is practically reduced to zero;" for otherwise he would have noticed that we could not possibly assume that, with the experiment so arranged, the friction could gradually diminish and ultimately become *zero* anywhere except at the axis of rotation of the pallets. By giving this limiting value to the friction at any other point he practically admitted that the effect produced by shortening the arm beyond a definite point must be the converse of that caused by lengthening it beyond that point.

246.—The above experiments, taken in conjunction with the considerations which follow, will enable us to lay down some *practical rules* for determining the best length for the escapement arms of clocks and watches.

The useful effect of the impulse varies with the height of the inclined plane employed to transmit the motive force to the balance.

247.—Some modern authors have relied solely on the unvarying principle that what is gained in force is lost in velocity or space passed through, and conversely; and, remembering that by increasing the height of an impulse curve its rate of advance is diminished in a similar proportion, they have asserted that, neglecting friction, the extent of the lift is a matter of indifference; they further state that when friction is taken into consideration the tangential incline imparts the greatest amount of motion to the balance.

This latter assertion is entirely gratuitous and in direct contradiction to observed facts.

The fallacy which renders the above conclusions erroneous lies in the fact that the several mobiles are considered either when only commencing their motion or in a statical condition.

Under these circumstances the moderator of an escapement is, so to speak, in repose; it is of no service. In order that its effect may be felt it is first necessary that it be moving, in other words we can only regard it when in action, that is in a dynamical condition.

What, then, is a moderator of an escapement? It is merely a body moving in consequence of an impulse imparted to it, and retreating in front of another body; as this latter starts from a state of rest, it cannot overtake the moderator unless travelling with a greater velocity (**28**).

248.—Let A (fig. 20) be the half-shell of a cylinder rotating as in a watch that beats 18,000 vibrations in the hour, and let its arc of vibration be 270°.

The resistance occasioned by inertia, friction, and oil, prevents the escape-wheel from immediately attaining to a velocity in excess of that of the balance.

Assume for illustration that $c\,b$ represents the distance

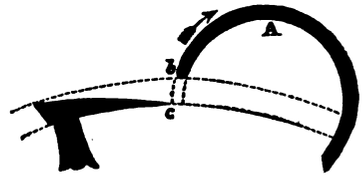

Fig. 20.

travelled by the point b during the extremely short period of time occupied by the wheel in traversing the arc contained between two consecutive teeth.

It is evident without demonstration that if the height of the inclined plane of the tooth be $c\,b$, this incline will follow the shell of the cylinder without pressing against it; this latter will retire in front of it, and the arc of 270° will rapidly decrease, finally remaining stationary at something materially less than 270°,

This circumstance did not escape F. Callet, the mathematician, as will be seen from a study of his memoir on escapements.

We thus conclude that in employing an inclined plane to transmit the action of a motor to an oscillating body, the energy of the impulse will increase and the arc of oscillation will become more and more extended if the height of the incline be gradually increased from zero up to a certain limit which we shall presently determine (**251**).

SIXTH EXPERIMENT.

249.—We have made the following experiments, which

anyone can easily repeat for himself, in order to practically demonstrate the truth of these theoretical deductions.

The escape wheel of a cylinder watch, in which the balance traversed a mean arc of 260°, was replaced by one with less inclined impulse curves, and we would specially mention that this latter had the advantage in being lighter and better made. With the new escape-wheel, however, although the friction was much less, the mean vibration of the balance was only between 210° and 220°.

SEVENTH EXPERIMENT.

250.—This is another form of the same experiment. In order to perform it we constructed an apparatus (fig. 5, plate III.), consisting of a lever A, mounted on pivots and balanced by a counterpoise weight *l*. At its opposite end an inclined plane *c* is centred on a pin *k* perpendicular to its surface. This plane is held between two plates of which only one *b* is visible; it can be fixed in any position by a clamping screw *f* working in a slot, and the inclination can therefore be varied as required.

The weight *y* is to impart motion to the lever.

A projecting steel strip *g* is fixed in the disc H, movable on a long pivot to which the pendulum P is attached. Through the action of this pendulum the disc oscillates in a manner analogous to the cylinder of a horizontal watch.

The experiment is conducted as follows, the apparatus being placed vertically:

The rounded corner *h* of the impulse curve is placed on the outer face of the strip *g*, and the bob P being held at a sufficient distance to the right by a fine thread, this thread is cut. The pendulum oscillates and the impulse curve acts on the edge of the strip *g* exactly as the inclined plane of an escape-wheel tooth acts on the edge of a cylinder when this latter is in motion through an impulse previously imparted to it.

In the course of the experiments we adopted the device of covering the rubbing face of the incline with a layer of lamp-black.

If the motion of the pendulum be *sufficiently rapid* and the inclination of the plane but slight, the entire length of the impulse curve may pass in front of the edge without touching it.

When the inclination was a little greater only the heel of the incline rubbed against the lip. The exact length of the

portion of the plane that came in contact with this edge was clearly shown by the removal of the black layer.

As the angle was gradually increased the line of black so removed was seen to extend from the heel towards the point of the plane until at a moderate inclination the plane was observed to rub throughout its entire length.

It will thus be seen that practice as well as theory, and with an equal amount of authority, shows how great an error is made by those watchmakers who pretend that, neglecting friction, the exact inclination of the impulse plane is a matter of indifference. Another proof of the correctness of the proposition above laid down is afforded by the following article.

There is only one degree of inclination for the impulse plane by which the maximum movement can be secured with the greatest regularity.

251.—We have just shown that if the inclination of the impulse curve be increased from a very small amount, the length of the surface that comes in contact with the cylinder edge becomes gradually greater until a certain height of the incline is reached when this contact takes place along its entire length.

We will for the present ignore these results; it will be easy by means of the theoretical considerations contained in the Introduction to demonstrate that the energy of impulse varies directly with the length of the impulse curve that comes into actual contact with the edge of the escapement arm, or very nearly so; and thus, that the maximum effect is obtained when the length of contact is the greatest possible, that is when it is equal to the length of the plane itself.

Consider the two triangles *a d* 6 and *c d* 2, or simply the inclines *a d*, *c d* (fig. 21).

The actual height of the plane is only a matter of indifference when the total mechanical effect is the same in the two cases. Here, however, this effect must differ materially.

An escape-wheel, when commencing its motion, must expend a very insignificant amount of force in neutralizing the resistance due to its own inertia and to oil, since it is essential, in order that this force may attain a maximum, that similar opposition due to oil and inertia be first overcome in the train itself. From this it follows that even although the space *a c* be traversed

out of contact with the edge of the escapement arm, the blow on the point *c* possesses but little energy, and it rather occasions a loss of force than any increment in the velocity with which the cylinder travels.

Even assuming that the impact performed any useful work,

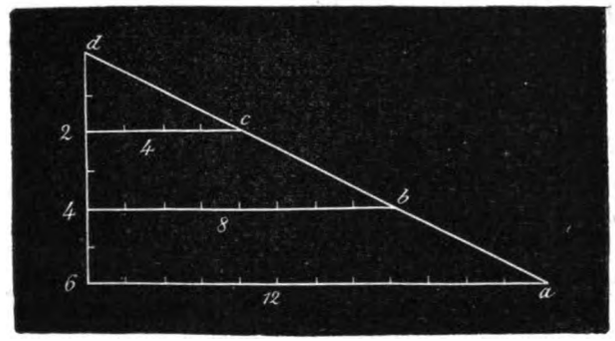

Fig. 21.

the small plane *c d* would still be required to travel with a velocity more than double that which it possesses.

So many and various are the resistances opposed to such an acceleration in its movement that one could not, even provisionally, admit it to be possible; these are: (1) the elastic reaction of the impact; (2) the radius of the wheel being greater at *c* than it is at *a*; (3) the reaction of the balance and balance-spring which could only double their velocity of motion if influenced by a far greater impelling force; and (4) the smallness of the interval *c d* which, after deducting an extremely brief period of rest due to the impact, does not give the wheel time to attain to any considerable velocity.

The instrument shown in fig. 5, plate III., will serve to practically demonstrate this fact. It must for this purpose work in a horizontal plane, the pendulum P being replaced by a balance E E provided with a balance-spring of moderate strength. If this experiment be performed with the pendulum the slight variations that occur are difficult to detect.

252.—It may then be considered as theoretically and practically proved that the extent of the oscillatory movement of the moderator will increase rapidly (providing the impelling force is sufficient and remains constant) as we gradually elevate the impulse curve from zero to a height which has to be determined.

Disadvantages of too great an elevation of the incline.

253.—It remains to be shown, before the demonstration can be regarded as complete, that beyond this limit the amplitude of oscillation does not increase in proportion as we increase the lifting angle; on the contrary, it slowly decreases and ultimately the motion would entirely cease if the height of the impulse curve were made excessive.

Let R (fig. 6, plate III.) represent the resistance opposed to the motion of this plane; *b* R its length, and *b a* the least height with which contact occurs throughout the *entire length b* R.

Draw the planes *g* R, *p* R such that they are respectively inclined twice and thrice as much as *b* R.

These heights will give a measure of the amounts by which the point R, representing the edge of a semi-cylindrical shell capable of a rotatory movement about its axis, is displaced; in each case this centre of rotation is situated on the horizontal line passing through the middle of the inclined plane under consideration.

The angles *n*, *t*, *v*, represent the angles of lift corresponding to an entire motion of translation of each plane in the direction indicated by the arrow.

We may now calculate the force exerted by each incline by dividing its base by its height (**134**); this gives:—

For the plane *b* R, $\frac{2}{4}\frac{5}{5} = 5{\cdot}5$ or 55
" " *g* R, $\frac{2}{9}\frac{3}{5} = 2{\cdot}5$ or 25
" " *p* R, $\frac{2}{13}\frac{1}{5} = 1{\cdot}5$ or 15

It thus appears that by applying to each a force of 1 they would be enabled to overcome resistances in the proportion of 55 : 25 : 15.

We see, then, that if, when the plane R *b* is employed, this force of 1 is capable of raising the body R to its full height, double that force will be required to produce a similar effect when R *g* is employed, and, with the plane R *p*, three times its amount. The resistances that have to be overcome are considerably greater than these values indicate when friction, which we have hitherto neglected, is taken into account, as it absorbs a gradually increasing proportion of the motive force.

254.—But every increase in the force brings with it various sources of irregularity due to excessive friction and the wear of rubbing surfaces, since the energy with which these are pressed

together is too great in comparison with the resistance opposed by the material of which they are formed.

These mechanical truths can be demonstrated by means of the instrument already described and represented in fig. 5, plate III.

The incline which makes contact throughout its entire length transmits the force with a maximum regularity.

255.—We have now proved that:

(1). Those impulse curves which are only effective over a portion of their surface do not impel the balance with an energy proportionate to the motive force applied, and they give rise to impacts (**46**); (2) an inclination in excess of the least by which contact throughout the entire length is secured, in future called the *mean incline*, occasions friction and a resolution of force of gradually increasing intensity (**67**).

In one case, then, there is a deficient impulse and disturbing impacts, and in the other, excessive force and pressures of a destructive nature with no counteracting advantage. It is thus evident, without any further demonstration, that by adopting the mean incline we secure a maximum of regularity in the impulses communicated to the moderator with a minimum of wear of the rubbing surfaces.

To determine the height of the impulse curve.

256.—If the several actions constituting a lift of the escapement could be mathematically considered in an exhaustive manner we should doubtless be able to accurately prescribe the height of inclined plane that is best adapted to any given escapement. But such a work would involve serious difficulties; it would be essential that certain data, the friction for example, should be accurately ascertained, and these at present are unknown to us, and would require for their determination experiments of extreme delicacy. The subject, moreover, could only be efficiently considered by scientific men of a high order, and the few who exist are not much encouraged to undertake so laborious a research by the uncomplimentary terms in which they are referred to in certain writings on Horology.

Hence, if such a work ever is undertaken, it will probably be long before it is accomplished. Fortunately for our art experience coupled with observation affords us a means of supplying this want sufficiently for all practical purposes.

257.—The entire arc of oscillation of a balance consists of:

(1) the lifting arc, and (2) the supplementary arc (94). Retaining the motive force constant, cause the amount of lift to vary by gradually increasing the inclination of the impulse plane; commencing with a very slight inclination we should find, if we properly applied the laws of Mechanics, that the supplementary arcs increase more rapidly than the lifting arcs before we reach the mean incline, and, beyond this point, the reverse is the case. We thus arrive at this remarkable result, that the mean incline gives the maximum supplementary arc, and, as a natural consequence, the most certain timing.

Besides the above we can resort to a mechanical means for determining this height of the incline; and we can deduce from it a method for ascertaining the requisite motive force and the height of the incline, as soon as the entire arc of oscillation and the number of beats per hour are fixed upon.

EIGHTH EXPERIMENT.

258.—We employed several escape-wheels that could be fitted on to one and the same pinion. The chord of the impulse curve passed through the centre of the cylinder.

The following were the results obtained:

1st wheel, lift 60° (35° on one side)—arc of oscillation 266°.
2nd „ „ 40° (24° „ „)— „ „ 255°.

The proportions:—35 : 266 : : 1 : x = 7·6
24 : 255 : : 1 : x = 10·6

give the following ratios:

1st wheel.—The lifting arc is to the total arc as 1 : 7·6
2nd „ „ „ „ 1 : 10·6

and, deducting the complete lifting arc from the arc of oscillation, we have:—

1st wheel—supplementary arc 206°.
2nd „ „ „ 215°.

NINTH EXPERIMENT.

259.—Two other wheels were now employed. They were of equal diameter, so that, as long as the distance apart of the centres remained constant, the total lift in the two cases was the same, notwithstanding the fact that the slope of the incline in wheel No. 4 was less than that of No. 3.

3rd wheel, total lift 40°, arc of oscillation 245°.
4th „ „ „ 40°, „ „ 220°.

and, deducting the lifting arcs from the arcs of oscillation,

3rd wheel—supplementary arc 205°.

4th „ „ „ 180°.

By slightly increasing the distance between the centres, so that the impulse plane of wheel No. 4 cut through the centre of the cylinder, the total lift could be reduced by 6° or 10° without the extent of vibration being sensibly affected, and the supplementary arc remained between 185° and 190°.

It is essential in performing experiments of this latter nature that the motive force be absolutely invariable, for every change in this force involves a variation in the height of the plane; the amount of force and the dimensions usually met with should therefore be employed. The introduction of masses and forces much out of proportion might occasion error.

Any change in the dimensions of the Moderator involves an alteration of the ratio between the impelling force and the pressure on the resting surfaces.

260.—Take an accurately regulated escapement provided with a pendulum beating half seconds, and replace this successively by others beating seconds and quarter seconds. From such a change another necessarily follows; the velocity of the wheel will be half the original amount with the longer pendulum, and twice as rapid with the shorter one.

The friction on the locking surfaces remains the same, while the energy of the impulse varies considerably in consequence of the changes in the velocity of the wheel (**28**).

Isochronism, which was secured with the intermediate pendulum, ceases to exist with the other two; and the disproportion existing between the pressure on the locking surface and the energy of impulse can be easily observed by the great variations in the rate of the escapement as the motive force is gradually increased.

So long as the amount of lift remains unaltered, these variations can only be avoided by modifying the radius of rest in accordance with the change in the energy of impulse.

261.—If we make a purely logical deduction from the laws of mechanics (**99** to **104**), this circumstance will merely enable us to conclude that *the existence of a certain relation between the length of the escapement arms and the dimensions of the moderator is one of the conditions of good timing;* we are compelled only to

adopt these general terms by our unvarying practice of supporting each proposition by an appropriate experiment.

262.—All escapements are subject to this law as regards the proportioning of their several parts; but it is specially important when employing the annular balance to take account of: (1) The mode of action of the balance-spring, for it differs from the pendulum in nearly always making the long arcs of vibration quicker than the short arcs; (2) the change which any variation in the weight of the moderator occasions in the friction of the pivots.

TENTH EXPERIMENT.

263.—This experiment is due to M. Henri Robert, and is described in Moinet's Treatise: "I constructed a clock with very great care, and it was so arranged that four different escapements could be adapted to it without involving any other change whatever. I will only describe the results obtained by employing alternately two forces in the proportion of 2 to 3, and from these experiments it will be seen that *the greater the length of the escapement arms in relation to the length of the pendulum, the more marked is the effect which differences in the motive force have on the period of the oscillations.*"

The author gave a table of his experiments in his work entitled *Études sur diverses questions d'horlogerie.*

ELEVENTH EXPERIMENT.

264.—The aim of this and the following experiment was to show clearly the relation between the two terms of the law we are considering when applied to the escapement with a pendulum, or with an annular balance.

I arranged a small clock train in the manner shown in figure 7, plate III. The movement is enclosed in a box C C. The axis of the centre wheel is prolonged downwards, and provided with a pulley P on which is coiled a cord *f* carrying a small driving weight M, hooked on so that it can be removed when requisite. The cylinder escapement is fixed on the upper plate, the circular balance being replaced by a small bar 1, 1, of rectangular section, terminated at its extremities by two small hinged arms carrying heavy masses *b*, *b*.

The object of such an arrangement will be manifest; by bending the arms inwards, so that the heavy masses approach the centre of rotation of the cylinder, exactly the same effect is

produced as if balances of different diameters, but constant weight, were successively employed.

When the train is in action, the finger *a*, fixed on the axis of the centre wheel, either traverses a dial, or starts from an initial mark; four complete revolutions are equivalent to 8,815 vibrations of the balance.

The cord was wound on to the pulley, and a weight of 7·5 grammes attached to its extremity; the 8,815 vibrations were accomplished

In 34 minutes 53 seconds when the heavy masses were at 1, 1;
„ 42 „ 38 „ „ „ „ 2, 2.

Similar experiments were made employing a driving weight of 30 grammes, and the 8,815 vibrations occupied

34 minutes 44 seconds with the heavy masses at 1, 1;
42 „ 35 „ „ „ „ 2, 2.

It should be noted that with the smaller balance the arc of vibration was 10° greater than with the larger one when the motive force of 7·5 grammes was employed, and the same was the case with the force of 30 grammes.

The above results show that by applying four times the impelling force, a variation of 3 seconds occurred with the larger balance, and as much as 9 seconds with the smaller. The sensibility of the escapement, in fact, in this latter case, became considerable, so that it had but little effect in counteracting any inequalities in the motive force. Now, since nothing was changed in the escapement except the size of the balance, this negative result can evidently only be due to the change from the relation initially existing between the lever that receives and transmits the impelling force, and the moment of inertia of the balance; this lever performs the double function of communicating motion to the balance, and then, in a measure, opposing its motion (**103**).

We shall subsequently revert to these experiments, and, at present, only consider them in so far as they are directly applicable to the subject we are discussing.

TWELFTH EXPERIMENT.

265.—The part taken by the balance-spring in the experiment above described is of the highest importance, for to it is due the gain which occurs during the longer arcs occasioned by the superior force. Were it not for the balance-spring a retardation would occur on increasing the amplitude of the oscillations.

In order to demonstrate this experimentally, I removed the balance-spring, as well as one of the heavy masses. The movement was then placed at such an inclination that the heavy mass remaining acted the part of a small pendulum, and its action was substituted for that of the spring. I could thus obtain, as required, pendulums of the same weight but different lengths by merely changing the position of the mass.

With a driving weight of 5 grammes, the two pendulums completed 6,611 vibrations as follows:—

Short Pendulum, in 14 m. 9½ s. Arc of oscillation 120°.
Long ,, ,, 19 m. 30 s. ,, ,, 100°.

And with the motive force increased to 15 grammes:—

Short Pendulum, in 17 m. 53½ s. Arc of oscillation 200°.
Long ,, ,, 22 m. 40 s. ,, ,, 185°.

The retardation caused by increasing the energy of impulse was 3 minutes 10 seconds in the case of the longer pendulum, and as much as 3 minutes 44 seconds with the shorter. The excessive loss in the latter case, being due to the pressure on the locking surfaces, clearly indicates that the radius of rest is relatively much too long for this short pendulum, or, to use the workshop term, the cylinder is too large for its balance.

We shall deduce some further interesting and novel conclusions from these experiments; they will be given in their proper place.

The size of the Escape-wheel is not a matter of indifference.

266.—The dimensions of the escape-wheel cannot be arbitrarily decided upon; they are a function of the velocity which the wheel itself can acquire, the effective height of the impulse plane (**251**), and the amount of friction on the lifting surface.

Let us consider an escape-wheel whose dimensions are double those of another wheel; and assume the smaller of these to be of a convenient size.

Two cases immediately present themselves; the larger wheel may have the same number of teeth as the smaller or twice that number.

In the first case, the two levers, namely, the power lever or the radius of the wheel and the resistance lever or the arm of the escapement, are magnified in the same proportions, so that the ratio remains constant; thus all the conditions are the

same as in the original case except that the resistances due to inertia and friction are altered.

When all the dimensions are doubled the mass will be magnified eight times, and the resistance due to inertia will increase in consequence. The friction of the pivots would become greater with every increase of the pressure and the amount of friction on the lifting surface must be from two to four times as great. The influence of the oil would become more sensible.

There will thus be produced a want of promptitude in the commencement of the wheel's motion, a necessity for increasing the motive power, and of modifying the height of the impulse plane; and, lastly, all the inconveniences already enumerated in the articles on resistance and elsewhere (**254**).

267.—In the second case, where the dimensions of only the wheel itself are doubled (the original wheel being assumed of a convenient size), the resistances opposed to its motion will increase in the manner indicated above, and, moreover, its angular path is reduced one half, the impelling force also being diminished in proportion as the lever arm is increased. Such a wheel will not only fail to act on the lifting edge throughout its entire length, but it will never have sufficient time to attain its maximum velocity (**28**).

This large wheel will be incapable of performing the work previously done by the smaller one, which it replaces, except by the application of an additional amount of motive force, and this excess of force introduces, as we well know, a whole string of causes of irregularity and wear. And this is easily understood when it is remembered that in order to double instantaneously the velocity of a moving body it is necessary to employ four times as great a force.

268.—The size of the wheel, then, is closely related to the velocity which any point on its circumference can acquire during the lifting action; that is to say, the size must be regulated by the velocity possessed by the moderator, since this velocity gives a direct measure of the height of the impulse curve. The dimensions of the escape-wheel, then, must vary with the nature of the escapement and the number of oscillations performed by it in a given time.

It is quite impossible to calculate what this proportion should be, but we can determine it or, at any rate, ascertain its

approximate value by experiment, taking the following condition as a basis; diminish the diameter of the escape-wheel, if a more rapid and energetic action is required, and increase it in a contrary case.

We shall subsequently explain the practical means which it seems advisable to adopt in order to determine the size of the wheel.

Important Observations:—Conclusion.

269.—The foregoing considerations appear to justify the following conclusion; success in *timing* in escapements with frictional rest depends on the inter-relation of all the parts, on the securing of a proper proportion between the moment of inertia, the impelling force, and the length of the radius of rest, or, in other words, of a kind of equilibrium between the powers and the resistances, the former only exceeding the latter by the very minute amount of energy that is lost at each oscillation of the balance.

270.—The problem appears at first sight to be simpler than it really is. It is not merely necessary to arrange the mechanism so that it shall maintain its rate of timekeeping under given initial conditions, it must be reliable in the future, that is to say, it must neutralize the causes of irregularity which time introduces.

The motive force communicated to the escapement gradually diminishes through the thickening of the oil and the wear of surfaces that are not so formed as to resist it; and, concurrently with this weakening of the force, the resistance at the locking surfaces increases until the proportion which originally existed between the impulse and the correcting pressure is entirely destroyed. Why does a recently cleaned watch go? Clearly because the power in action is greater than the resistance; and why does a watch stop when the oil has become thick, if it is not that the resistance on the locking surfaces has so increased as to be greater than the impelling force?

One element in an escapement, namely friction, varies with the material employed and the degree of polish given to it, as well as with time, the nature of the oil, and the extent to which it is decomposed. Changes in the condition of rubbing surfaces (for example, in one arm of the escapement as compared with the other) may exert a contrary influence.

271.—The mode of suspension of a pendulum requires to be carefully considered, for the support will exert an amount of influence on the rate dependent on its greater or less freedom.

The balance-spring will be subsequently discussed. It will be seen to afford us a most complete solution, so far as regards the escapements of watches, of the problem whose elements we have been discussing.

272.—In every escapement the lifting action is of primary importance. Practical men should thoroughly acquaint themselves with the principles laid down in the Introduction (**94** to **104**), for it is essential that the lift take place under conditions which are as favourable as possible to the timing and the accuracy of the several actions, and involve a minimum of impact. Each impact of appreciable energy is liable to derange or render variable some of the effects, in addition to entailing the inconveniences already pointed out.

A body at the commencement of its movement will, if not impeded by any obstacle, travel for a certain period with a uniformly accelerated motion. Were such the case during the lift, the escape-wheel would fall on to the locking surface when travelling with considerable velocity, and the impact against the locking surface would destroy some of the energy of the balance; the exact amount so destroyed would become greater as the velocity of the wheel was increased.

It is advisable then to ascertain for the given conditions which of the three kinds of motion, uniformly *accelerated*, *uniform* or uniformly *retarded*, can be employed during the lift with the greatest advantage.

A uniformly retarded lifting motion, when equal arcs are traversed by the balance in equal times, generally, in consequence of the acceleration in the velocity of the balance during the lift, corresponds to a uniform angular movement of the wheel.

RULES FOR CONSTRUCTING THE CYLINDER ESCAPEMENT, DEDUCED FROM THEORY AND EXPERIENCE.

The Form of the Teeth.

273.—It has been shown by extracts from Jodin's work (**199**), the very guarded note by Moinet (**207**), and by the general considerations which precede article **211**, that the

determination of the best form to be given to the impulse plane has made no progress for the last fifty years, and that the suggested improvements have been no more than old blunders advanced as though they were novelties; discussed, subjected to the test of experiment, and finally decided upon long ago by Jodin.

When ignorant men officiously put forward their notions as laws it matters little; but it is much to be regretted when authors of ability, ignoring that safe guide, experience, assert the results of childish speculation to be rules of practice; the most they can do is to mislead young apprentices and discourage studious workmen by causing doubts in their minds as to the use of science.

Straight and Curved Inclines.

274.—It is impossible to give an answer to the question, "What is the most advantageous form for the incline in a Cylinder Escapement?" until the properties, good and bad, of every possible form have been shown by the help of theoretical laws corroborated by carefully conducted experiments.

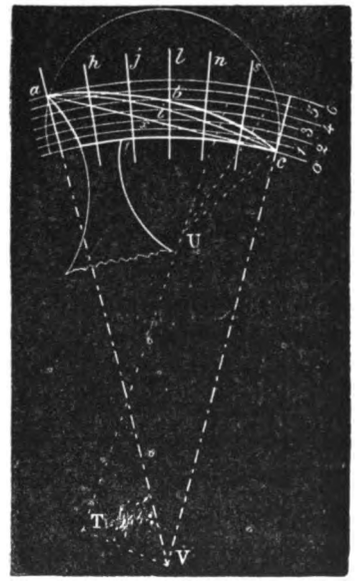

Fig. 22.

275.—If the three forms of incline, (1) rectilinear, (2) a curve described with the radius of the wheel, (3) a greater curvature at the commencement than towards the end of the incline, be studied in the manner already indicated, it will be found that:

276.—When the radius of curvature of the impulse curve is the same as that of the wheel, *a i c* (fig. 22), the angular movement of this wheel corresponds very approximately with that of the balance. But the motion of the balance under the pressure of the wheel is gradually accelerated, whence it follows that the motion of this wheel itself can be considered to be *uniformly accelerated.*

277.—When the incline is straight, *a x c*, the angular displacement of the wheel becomes gradually less than that of the balance, but since the balance accelerates its movement, as we have already remarked, it naturally follows that the motion of the wheel is *very nearly uniform*, for it loses on the cylinder, and this is known to be accelerated.

278.—Lastly, when the incline is much curved at its point, *a b c*, the angular displacement of the tooth is at first very slight, but increases with great rapidity. It traverses its effective path with a *gradually increased acceleration* and, completing the latter portion of the lift with great rapidity, falls with violence on to the locking surface. These impacts are easily detected even by an unpractised ear.*

Inclines of Pronounced Curvature.

279.—An incline having a face of pronounced curvature, especially if this be towards the point of the tooth as represented by the arc *a b c* (fig. 22), is absolutely useless.

As we have already seen, when the tooth is shaped thus, it is almost stopped immediately on commencing its motion through its very short angular path, and only attains a maximum velocity towards the end of the lift. A great portion of this lift then is nothing more than a sliding action, or, rather, a very long drop; its most obvious effect is to disarrange the entire system by the impact that occurs (**46**), and to absorb a portion of the *vis viva* of the balance, thus diminishing the extent of the oscillation (**272**).

With the form of tooth now under discussion a part of the

* The lines *s n l j h* (fig. 22), divide the space traversed by the tooth during a lift into six equal parts; the arcs 1, 2, 3, 4, 5, divide the displacement of the edge of the cylinder into the same number of equal parts. The figure shows that the curve *a i c*, having a radius equal to that of the wheel, passes through the points at which *s* cuts the line 1, *n* the line 2, and so on.

As to the curve *a b c one-sixth* of its displacement *c s*, corresponds to *two-sixths* of the displacement of the cylinder, 0 to 2; at first it moves with half the rapidity of the cylinder, but it very soon commences to increase its velocity. The reverse is the case with the straight plane *a x c*.

length of the incline simply glides past the lip, and the blow on the locking surface is so violent that, with some escapements, the heel of the tooth rebounds against the shell, producing a noise analogous to that which is occasioned by the coils of the balance-spring coming in contact with each other. This effect may be prevented by slightly working down the planes of the teeth.

EXPERIMENT.

280.—Many Geneva watches, though well made and of sufficient thickness, are very liable to bank, even when the original mainspring has been replaced by one much weaker, and giving six turns. In such a case it is only necessary to so work down the escape-wheel teeth as to slightly remove the heel end of the inclined plane, and banking will be prevented. For such an operation has the effect of changing the incline into one which is more curved towards the point, and flattened towards the heel; hence there will be a greater velocity acquired when the drop occurs, and, as a consequence, a diminution in the amplitude of the oscillation of the balance, for it is thus deprived of some of the energy it had acquired.

We shall, in the sequel, give the results of further experiments, but we must at once deprecate entirely the use of a curve of pronounced convexity, and we can confidently assert, having made the experiment to our cost, that watches constructed according to the principles which are held by the advocates of curves of this form are, as a rule, very bad timekeepers.

Inclines which are straight or very slightly curved.

281.—The only curve that can be adopted with any advantage is the one described with the radius of the wheel itself (**276**).

We will give, side by side, the properties of this curve and of a straight plane, but before doing so must here make a correction. It has reference to a point which, trusting to certain authors, we assumed, in the first edition of this work, to be proved.

It is there stated that: "In the cylinder escapement, when the tooth leaves the locking surface, the wheel, in virtue of the laws of inertia, is incapable of commencing its motion instantaneously. There is an imperceptible period of stoppage, and, since the cylinder is travelling with considerable velocity (18,000 vibrations an hour), the tooth does not touch the edge with its point, but with a part of its surface rather more distant from the axis of rotation of the wheel.

"It may be conceived that with a straight incline the space traversed before contact is greater than when it is curved; and this truth may be easily shown by means of a figure. Assume that, at the moment when they should touch, the edge has reached the position *a* (fig. 23). The point *o* on the curved incline and *r* on the straight one will be the points at which contact occurs."

282.—The first observation of these authors is legitimate, but it requires modification; it is always confirmed when examined by means of the heavy instruments used for demonstration, in which a certain mass of matter is set in motion, but in the escapements of watches, if the surface of the incline be

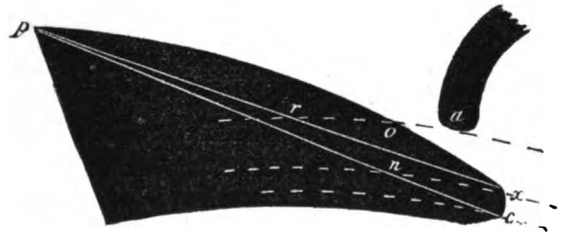

Fig. 23.

covered with a thin layer of a substance such as lampblack, it is very difficult and generally impossible to prove that any kind of plane could pass without contact taking place.

This unexpected result, very often confirmed, would be extremely awkward, but a little reflection will soon show the cause of the apparent contradiction.

283.—An examination of several cylinder escape-wheels will at once show that the points of the teeth are not sharp, but that they are terminated by small arcs of circles as seen at *x c* (fig. 23).

The radius of curvature of this portion will be greater in comparison to the length of the incline as the dimensions of the escapement become less.

Contact with the locking surfaces occurs at *c*, and we will assume that, in consequence of the resistance caused by inertia, oil, &c., the space traversed by the tooth without influencing the cylinder is equal to a tenth of the length of the incline or else to the arc *x c*, the wheel being entirely unconstrained. It necessarily follows that the first contact of the plane with the lip will take place at the point *n* if the tooth has a very sharp point and the plane be straight, and at the point *x*, if the

extremity of the tooth is rounded. In the latter case the distance between x and c on the tooth of such a small wheel is so very slight as to be almost inappreciable even with a powerful eye-glass.

This method of verifying will only enable the observer to clearly detect the differences, in the case of certain kinds of watches, when the angle of the inclined plane is very acute.

Properties of the straight incline.

284.—The advocates of a straight incline, when comparing it with the curved form, claim for it the following advantages:

It occasions less friction since the rubbing surface is shorter;

The resolution of the force is identically the same at each point;

Lastly, it occasions a greater motion of the balance.

Of all these advantages claimed for it, none of which they support by experiment, only one is real, and that is the last.

285.—The resolution of the force does not remain precisely the same at each point; and, as regards the comparison of the total amounts of friction with the straight and the curved incline, if these are made to depend merely on the length of the rubbing face, the conclusion arrived at can only be false, for it is based on a wrong principle.

In truth, what we must consider is not an infinitely small difference of length, which is quite beyond our powers of measuring, but the *intensity of the friction* that takes place against the cylinder edges. The straight incline exerts a more energetic pressure, and, since all friction is proportional to the pressure (**38**), it follows that, so far from the curved surface causing an excessive friction, it is, on the contrary, the one which occasions least.

286.—We have already seen that the wheel with straight inclines moves slower during the latter portion of the lift, and thus the impact on the locking surfaces is less. Greater pressure on the cylinder edges, a less amount of drop; these are more than sufficient to account for the greater extent of the vibration secured by the straight incline.

Properties of the curved incline.

287.—The advocates of an incline of such a curvature that each equal portion causes the balance to describe an arc of equal extent, say:

It obviates setting during the act of winding up the watch;

The lift takes place with greater regularity;

Less resistance is opposed by the balance-spring towards the end of the lift.

All these advantages are utterly illusory; the one point of importance is omitted.

288.—There is no likelihood of a modern cylinder escapement, if well made, setting at the moment of winding; hence it is quite unnecessary to take precautions against a fault which does not exist.

The regularity of lift, so far from being beneficial, is a disadvantage, since it can only be obtained by accelerating the angular movement of the wheel (**276**) and increasing the drop.

With regard to the increased resistance of the balance-spring, this is only an unproved hypothesis that has originated in the persistency with which some authors regard this spring just as it is abandoning a state of rest and not when travelling in virtue of a rapid movement already acquired. We shall have to revert to this subject.

One test, which would be crucial, of the existence of the several qualities claimed as belonging to the curved incline we are now considering, would be that it favoured the extension of the supplementary arc; but as a matter of fact just the *contrary effect* is observed.

289.—The real practical advantage of the curved incline is that the lift occurs under a but slight pressure, and the friction is more evenly distributed. We must be permitted the use of this latter expression, as we are unable to find one that better conveys our meaning.

RÉSUMÉ.

290.—With a curved incline then:—harsh drop; slight pressure on the cylinder edges.

With the straight incline:—slight drop; excessive pressure on the edges.

It necessarily follows that, when there is identity in every other particular:

The *straight incline* gives somewhat longer supplementary arcs, but the wear of the edges will be rather more rapid.

The *curved incline* will ensure the edges resisting wear for a greater period of time, but will give rise to a loss of some degrees in the amplitude of the oscillation.

These theoretical conclusions, which are strictly verified in practice, indicate most clearly the one true solution. It is this:

291.—*The impulse curve should have a very slight curvature* intermediate between the straight line and the curve we have been considering.

When employing this incline of slight curvature the extent of the oscillation will be approximately equal to that which results from the action of a straight plane and, the friction on the lips being somewhat less, the rubbing surfaces will resist wear longer.

292.—It is not necessary to consider the wear that may occur on the locking surfaces; there has hardly ever been a case of a well-made escapement wearing at this point. The edges are always the first to suffer.

We have said nothing with reference to the slightly concave incline, suggested by some authors; it would only hasten the destruction of the edges of the cylinder, and may therefore be disregarded.

293.—As we shall prove by the experiments described below, theory and practice are entirely in harmony as to this question. But we would first draw attention to the immense amount of time that watchmakers have wasted in discussions at once idle and fruitless, which too often tended not to set forth principles but to make a parade of their own self-love, since the conclusion we have come to, the only true one, is exactly the solution arrived at by Jodin and formulated by him more than a hundred years ago!

EXPERIMENTS.

294.—"In questions of this nature," says Moinet, "argument may very easily mislead, and it is therefore safer to resort to experiment." We have followed the sound advice of this author, and the results at which we have arrived fully confirm the accuracy of the above theoretical deductions.

We would beg our confrères to do as we have done. Then will come to an end the useless discussions that have been revived almost annually for the last hundred years, and must be interminable, as they depend on the exact use of language or on so-called mechanical principles which, however, science does not recognize.

First Experiment.

295.—A cylinder escape-wheel, constructed with care and accuracy, was fitted to an escape-wheel pinion. The curve of the incline was very nearly that usually adopted at the present day; the lift was as much as 60° and the balance had a mean oscillation of 260°.

After these preliminary observations had been made the wheel was removed, and the curved inclines reduced and polished with very great care in such a manner that the extreme points of the plane were in no case touched, the wheel being then restored to its position in the watch. On winding this up the mean oscillation was found to be from 266° to 270°; that is, from 6° to 10° more than when the curved planes were employed.

Second Experiment.

296.—This was precisely the same as the one just described, except that another wheel giving a lift of 40° was employed.

The result was similar, but the differences were somewhat greater. This we consider to have been due to the fact that the friction on the cylinder edges was slightly reduced (**253**).

The initial arc of oscillation of 245° was raised to between 260° and 265° by employing the wheel with its inclines reduced; that is, there was a gain of from 15° to 20° by using this latter.

Third Experiment.

297.—With a view to render these practical demonstrations more complete, and to supply an answer to an objection which, though not serious, might be raised, namely, that the weight of the wheel was altered by the reducing of the teeth, we employed in this experiment the apparatus described at page 133, and represented in fig. 5, plate III.

To the extremity of the lever A were fixed two inclines at a short distance apart, and in planes parallel to each other. One of them, *c*, was straight, and the other, *d*, had a curved face. They were rigidly connected together, and could be caused to rotate round a centre *h*.

They had two points in common, namely the points and heels which were directly superposed, so that the total lift was the same whichever acted on the strip *g*. This strip, it should be noticed, is provided with a slot, and thus the incline *d* is allowed to pass without contact when *c* acts on the edge of the strip; and, when the incline *d* is required to act on the edge, it

only becomes necessary to depress the whole system E H E, by means of a small adjusting screw placed under the instrument. The incline *d* can then act on the edge of the strip, while the plane *c* passes above it.

A small spring or pawl *z*, is screwed on the arm A in such a manner as to rest against the rim of the disc H immediately the impulse is completed. By easing the screw, *z* can be slightly rotated so as not to come in contact with the disc.

298.—The apparatus when thus arranged is available for three classes of experiment.

1. To ascertain the impelling force of each form of incline on a body which turns freely on its axis. (In this case *z* must be turned aside.)

2. To determine the difference in the movement occasioned by the two planes when the impulse terminates with an impact, after which a continuous pressure is brought to bear on the moving body. This is precisely the case of a cylinder escapement when in action. (The spring *z* must be so placed as to press against the disc immediately on the plane quitting the edge of the strip *g*. The pressure on the rim will then always occur with the same radius of rest, and the rubbing surface will, in every case, be the same.)

3. To establish the relative values of the impacts on the locking surfaces with each form of impulse plane.

Care is necessary to conform as much as possible to the conditions actually met with in horology; if the forces or velocities are too great or too small, contradictory results may be obtained. The incline must in each case act throughout its entire length.

299.—The mode of action of the instrument has been already explained at page 133. Every experiment was repeated several times under conditions that were identical for each incline.

With the spring *z* brought into play so as to correspond to a locking action on the edge of the disc H, at the instant at which the heel of the incline escapes from the strip *g*, the greatest arc of oscillation was obtained by employing the straight incline *c*.

It is specially interesting to examine the energy of the drop, for it is found to vary with the incline employed; the plug *k*, sliding with friction, is adapted to the instrument for the

purpose of measuring this energy. If z be removed, the lever A falls on the head of this plug or piston as a support, and the space through which it is driven gives a measure of the impact.

From numerous experiments, although k was held by the spring with considerable firmness, we have ascertained that it is forced at least two or three millimetres lower with a curved than with a straight incline.

Concluding remarks on the Straight Incline.

300.—Berthoud employed a straight impulse plane in his marine chronometer No. 8, and this was the best he ever produced. As a modern writer has observed, and as is quite in keeping with Berthoud's character, he would never adopt a system that was in favour with his opponents unless thoroughly convinced that it possessed real advantages.

Jodin and Jurgensen assert that this form gives the greatest possible extent of vibration.

Lastly, of modern horologists, M. Henri Robert came to the same conclusion as the result of his experiments.

We might add to this list the names of several provincial watchmakers (for example, M. Foucher, of Bourges) who have studied the subject experimentally and satisfied themselves that straight inclines always secure a slight increase in the arc of oscillation.

We are then fully justified in considering the question as having been exhaustively discussed both theoretically and practically, and in retorting to those who dispute the facts: Experiment under proper conditions and then judge!

301.—Possessing as it does this immense advantage, which is arrived at both by pure reasoning and by practice, the straight incline would have triumphed had not the very same theoretical considerations which, as it were, brought to light the advantages of the form, indicated with equal clearness the objections to its use.

For we have seen that the pressure on the edges of the cylinder is greater with a straight than with a curved incline, and hence it follows that these edges, which as a rule are the only portions of the cylinder subject to wear, will withstand the pressure of the straight impulse plane for a shorter period; and this is precisely what the long experience of old watchjobbers has proved incontestably. But we would at the same time observe that, although with the commoner class of watches

worn or pitted cylinders are most frequently met with when the inclines are straight, it is only fair to remember that in the best workmanship, where the escapement is carefully made, perfect cylinders are often met with after they have worked against straight inclines for twenty years or even more.

It is as well to draw attention to this fact, which proves what all watchmakers know, namely, that it is possible to make excellent cylinder escapements where the inclines are straight, but to argue from such a fact that this form of plane secures a minimum of friction, and that manufacturers should abandon the slightly curved form at present in vogue is to draw two false conclusions from a single incomplete observation. Our readers doubtless now look at the matter in its true light, but in order finally to do away with an erroneous impression we will make a few further observations.

302.—Besides occasioning an increased pressure, the straight incline projects less from the pillar of the tooth, so that the oil is drawn away from it and retained by the pillar. To avoid such a fault of construction and cause of wear it becomes necessary to make the pillar very thin, or else to load the circumference of the wheel by making the teeth large; and both these extremes have objections.

303.—We have repaired a great number of old watches with straight inclines, and we would specially cite two which had been going uninterruptedly for forty-three and forty-eight years, and the continuous contact of the tooth had occasioned nothing but very slight marks on the edges of the cylinder.

Does this demonstrate the complete superiority of the straight incline? No, decidedly not; but we have endeavoured to ascertain the cause of this remarkable preservation of the frictional surfaces in many of the older cylinders, and it was neither long nor difficult of discovery.

The old cylinder escapements with a steel wheel were made by the very best workmen of the day, and they took the greatest possible care in selecting the metal and in the operation of hardening by which it is so easy to utterly ruin the steel. Their cylinders were made of hammered steel, while those of the present day are always of drawn steel; a very important difference, for the texture of the steel in the two cases differs considerably; the first kind requires a less degree of heat in hardening, and can be polished better and more evenly.

We should feel that we were doing an injustice to the common sense of competent watchmakers if we pointed out the conclusions to which the above comparison leads; conclusions that show clearly how great a mistake modern manufacturers would make were they to replace the slight curvature they have adopted by the rectilineal form of incline.

Is it necessary that the form of Incline bear any relation to the variable resistance of the Balance-spring?

304.—The attempt to give the incline such a form that the motive force shall be proportional to the gradually increasing resistance of the balance-spring is the result of a fallacy.

The fault which this so-called improvement professes to neutralize does not exist.

Those authors who regard the balance-spring as an impediment to the lifting action, make a mistake, because they consider it when at rest or starting from a condition of rest, and not when in a state of motion; and they forget that this spring, besides having opposed a considerable reaction to the force which wound it up, possesses, in virtue of the laws of elasticity and inertia, a tendency to continue in motion, as is the case with all moving bodies.

If we examine minutely the series of operations involved in the going of a watch, we shall soon perceive that at the commencement of the lift, the balance-spring assists the motion and impels the balance; but as soon as this latter has taken up the impulse, its velocity is in excess of that of the balance-spring, and it drags this along with it. The balance then is influenced by the balance-spring, but in opposite directions at the beginning and end of the lead; and further, the resistance of the balance-spring at the end of the lead depends solely on the action which the wheel exerts on the balance.

It seems evident, without a demonstration being necessary, that the greatest action of the motor, that is of the escape-wheel, should, if possible, occur at the commencement of the lift; this force, acting on the balance when less constrained, will communicate a greater velocity to it, and the balance, possessing an increased *momentum*, will acquire a greater energy; its oscillation therefore will be of greater extent.

The height of the Impulse Plane is the only exact measure of the Lift.

305.—As Moinet has observed, authors often do not agree on the subject of the lift; and this is hardly to be wondered at,

since some measure the total lift and others only the lift on one side, a quantity which is not exactly half of the total lift.

We must at the outset, then, carefully distinguish the *real lift* from the lift as measured by the methods ordinarily in use.

306.—THE REAL LIFT *is the arc described by any point on the balance during the passage of the entire length of the incline in contact with one edge of the cylinder, and this lift* (*a b* or *c i*, fig. 24) *when doubled gives the total real lift;* the only exact measure of the force exerted by a given impulse plane.

307.—THE APPARENT LIFT *is the circular arc enclosed between the two points marked on the plate* (0° and 40°) *and corresponding to the ends of the two semi-lifts.* These two points enable us to ascertain where each semi-lift terminates, but do not indicate the points at which they commence. The *central point*, however, placed between the two that indicate the extremities of the lifting arc, (*d*, fig. 24) gives us the means of ascertaining them. Every watchmaker must have observed

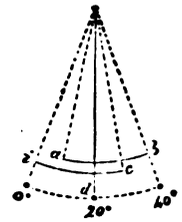

Fig. 24.

how rarely an escapement is met with whose wheel commences its movement, that is the *lift*, exactly when the mark on the balance is opposite the central point *d;* the wheel nearly always begins the lift 2°, 4°, 6°, or even 10° in advance. It is evident that these amounts should be included in each semi-lift to which they belong, and that if three escapements possess a lift of 40° between the two marks, but the semi-lifts commence respectively 2°, 5°, and 10° in advance of the centre, the period of action of the inclined plan is represented not by 40°, but by 44°, 50°, and 60°. Thus, every conceivable form of tooth, whatever be the inclination of its impulse curve, would give, providing the diameter of the wheel (measuring the heels of the planes) were constant, a constant total lift, although differing materially in impelling force.

308.—We thus see how great a mistake is committed by

those who advocate only one total lift (0° to 40°, fig. 24) for all escapements, and how it is that, in practice, we seldom meet with escapements of about the same dimensions which give a total lift of 40° and whose impelling forces differ materially.

309.—The *real* lift is invariable since it depends solely on the height of the plane; but the *apparent* lift varies with the opening of the cylinder, and also depends on its pitch with the escape-wheel. As has been already seen it is deceptive, often to the extent of nearly a third of the length of the impulse plane, thus ignoring the one thing that it is especially important to deduce from it, namely the impelling force as calculated from the height of the plane.

310.—Let it be for the future understood that the lift always means an entire passage of the incline in contact with one edge of the cylinder, and that in order to ascertain the *total real lift* it is only necessary to double the amount thus obtained.

Every watchmaker who fails to make this distinction between the two kinds of lift will be always liable to fall into serious error, and will be *perpetually deceiving himself as to the effect produced by a variation of the distance apart of the centres of rotation of the two mobiles* (**17**).

The chord of the arc of the incline should pass through the axis of the Cylinder.

The Drop.

311.—It has been shown in the Introduction (**46**) that whenever movement is transmitted, except in certain special cases, it is necessary to provide against impacts, which, in addition to occasioning a shake, cause both wear and a dissipation of force.

Every watchmaker has observed the effect of the drop on the extent of vibration, and how difficult it is to secure a permanent rate with an escapement in which the drops are excessive; this is especially the case when the outside drop is too great. This latter remark has not been made before by any author. It gives expression to a fact which skilful practical men cannot fail to recognize, and whose cause is clearly indicated by theory.

The outside drop has a greater effect on the going of the escapement, because (1) it occurs at the extremity of a longer arm of resistance; (2) the friction of the point of the tooth at the instant of contact is what is known as *engaging* friction

(**105**), and its resistance is therefore more marked than that of the internal friction. It is easy to perceive why the force exerted by this backward contact, as it may be called, becomes greater as the drop is increased. (These considerations will enable us to explain why a cylinder is always more deeply and more rapidly pitted on the external than on the internal locking surface.)

With a tangential escapement the fault would be less conspicuous since the friction would no longer be engaging; but it is useless to further discuss this case for it will be subsequently seen that with such an escapement it is impossible to secure vibrations of the requisite extent.

To insure that the drop is no more than sufficient to secure the proper action of the mechanism, it is of the first importance that the middle of a straight incline correspond to the centre of the cylinder. The external drop will be the greater, as it diverges in either direction from this point.

It will be seen that in the separating and bringing together of the mobiles, we shall have a practical means of ascertaining when the middle of the straight incline corresponds to the centre from the fact that it is the position in which the external drop is a minimum.

These considerations only apply to the case of a straight impulse plane, for it represents the diameter of the cylinder, and

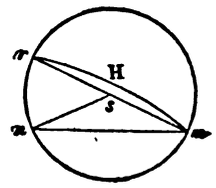

Fig. 25.

the curved incline is somewhat above it as shown at H (fig. 25).

312.—The older watchmakers adjusted the escapement so that the middle of the straight plane came rather beyond the centre of the cylinder, in order that the point of rest might be tangential; among modern makers it is universally recognized that more is lost by making the outside drop excessive than is gained by a slight diminution of the friction during rest.

Some watchmakers at the present day, who from insufficient knowledge are not in a position to judge correctly as to the cause of the circumstances which they observe, have asserted that they obtained a greater regularity by making the middle of the plane fall a little short of the centre of the cylinder.

This result often occurs with cylinders that are much closed, and it is in no way remarkable. It only proves two things: firstly, the ignorance of those who exalted into a law what is only a simple isolated fact; and, secondly, that the escapement had been badly proportioned, since it was found necessary to render the friction on the locking surface more harsh, in order to secure the most efficient proportion between the powers and the resistances. It is no more than a case of one fault being compensated by another.

313.—A series of experiments, conducted on a number of watches, have satisfied us that: (1) the escape-wheel tooth would occupy the most favourable position in the cylinder when the line drawn from the heel to the point (of contact) of the tooth passes through the centre of the cylinder; (2) if, when the tooth is thus placed, the vibration is constrained or of insufficient extent, this must be considered as due either to the escapement being badly made, or to its being ill-proportioned as a whole to the motive force communicated to it.

Except in certain few cases, which could be easily explained on other grounds, we have ascertained that by setting the plane beyond the diametral position, the extent of the vibration is hardly increased, even by the number of degrees added to the lifting arc, and the escapement is made more sensitive to changes in the consistency of the oil; and further, if this incline is set short of the centre (unless the cylinder be very slightly cut away) several degrees of oscillation are lost.

We only give here the indisputable results of our experiments; it would be impossible to give the minute details which their description would require, and this has, moreover, been done very fully in the *Revue Chronométrique*.

The height of Impulse Curve and the Lift.

314.—The angle between the base of the tooth and the inclined plane varies materially with different authors. Thus, while M. Wagner says it should only be 6°, it is nearly 10° according to Berthoud and Moinet, and finally 12° is recommended by Tavan.

These numbers correspond to a wheel of fifteen teeth, and they are inapplicable in the case of a wheel with either more or less teeth. If the angle of the tooth be in each case the same, a wheel with fewer teeth will give a greater lift, and a greater number will cause the lift to decrease.

Each of the above quantities represents half of the lift that occurs on one side, or a quarter of the total *real* lift*; we thus obtain for the entire *real* lift, 24°, 40°, 48°. We are then utterly embarrassed in our selection; one single definite value is needed, and three are presented varying from 24° to double this amount. According to one authority, when a lift of 12° on either side is exceeded, no corresponding increase occurs in the impulse communicated to the balance, and this can be easily *demonstrated* by the aid of geometry and the laws of the inclined plane. Another says *from practical experience* it appears that the inclination giving 24° of lift on either side "is the most convenient that can be adopted."

While endeavouring to ascertain the truth, we are at once met by a glaring contradiction between theory and practice; but be it observed that we have proved there to be no more want of harmony in this case than in a thousand other similar cases. The contradiction is only between the authors themselves, who, influenced by various prejudices, have each looked at only one aspect of the question, and have thus been led, one to apply erroneously a principle that is scrupulously exact, and the other to deduce a general rule from a practical result, which, though confirmed in most cases, is not so in all.

What has been said above on this subject (**247** and the following articles) should abundantly suffice; but as we are dealing with a false impression that is very deeply rooted, we will add some further considerations.

315.—Modern watchmakers, who are satisfied that it is necessary both to compel the point of repose to act tangentially to the cylinder, and at the same time to keep the centre of rotation of the cylinder on the chord of the impulse curve, never support their statement, which after all is only a personal opinion, by any experimental evidence whatever.

Amongst other mistakes, they assert that watchmakers have hitherto misconstrued all the laws of matter in not making the rest tangential.

One cannot agree to such a statement, unless, indeed, it be

* It is proved in geometry that the angle subtended by a given arc at the centre of a circle is double that subtended by the same arc at its circumference. Thus, if the angle *r s n* (fig 25) is 12°, the angle *r m n* subtended at the circumference by the same arc, *r n*, will be half this amount or 6°, and consequently the angle between the base and the incline of an escape-wheel tooth must equal one half the angle of lift (as measured at the centre) or, what amounts to the same thing, a quarter of the total real lift.

admitted that the artists who, for the most part eminent, have successively studied the cylinder escapement during the last century, have all been ignorant of the fact that, with the escapement tangential, the friction on the locking surfaces is least detrimental; and we must further suppose that they adopted a contrary principle blindly, without making any careful study of the subject. References to their own experiments, which are found scattered through their writings, are sufficient to prove that such was not the case, and that so far from undervaluing the tangential rest, it was precisely the position which they adopted from the very first. If they did abandon the system, it was only after mature consideration, and when experience had proved that its advantages did not counterbalance its objections.

316.—That the middle point of the chord of the plane, that is the line joining its point and heel, should not go beyond the centre of the cylinder, has been already demonstrated (**311**, etc.); but from this condition it necessarily follows that, in order for the escapement to be tangential, the angle between the impulse plane and the base should not exceed 6°.

Draw on a large scale the exact form of the tooth required for a tangential escapement (A, fig. 26), and side by side with it describe the same tooth (B), in accordance with the data given by practice. Every practical man would unhesitatingly

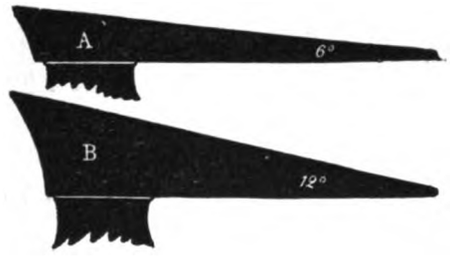

Fig. 26.

assert the impossibility of ensuring a good timing when the height of the teeth is as small as that represented at A.

Theory fully explains the causes of this observed fact (**247**, etc.). In connection with it we will refer to the following:

317.—Not so very long ago the watchmakers of Geneva concluded, from the theory then in favour, that an escapement would be improved by reducing the inclination of the impulse

plane to a minimum. The great majority of the watches of that period exhibit the error here referred to, and these same watches, while still having a lift of about 30°, have long been an annoyance to watchjobbers, who are good judges on such a matter. They are unanimous in asserting that as soon as the oil has lost its initial fluidity such watches are difficult to time, go very irregularly, and have short vibrations; and that, in consequence, they require frequent cleaning, very strong mainsprings, etc. Where, then, is the advantage in a slight inclination if it renders the watch a worse timekeeper, and requires a more powerful spring? And what after this can be expected of a tangential impulse plane?

All that has been said above is directly applicable to the callipers of watch actually in use, but the conditions enumerated in paragraph **320** must be taken into account.

Accelerating and Retarding Forces.

318.—The action of the wheel on the cylinder is, as we have shown, of a composite nature: during the lift, the motive force accelerates the movement already acquired by the balance; during the locking, this same force constrains the movement, checking it more and more.

In the first edition of this work, in order to simplify the study of escapements, we employed the terms, *accelerating force*, to indicate the power which is acting during the lift, and *retarding force*, for the pressure on the locking surfaces, and we propose to retain these expressions.

The extent of motion of the balance is closely dependent on the proportion existing between these two forces; and when the most convenient ratio has been secured with a given motive force, it is necessary to modify it in accordance with every increase or diminution in that motive force, however slight the change may be.

The reason for these results has been clearly shown from theoretical considerations (**103**, etc.) and we will here only add some experimental evidence in support of it.

Every watchmaker is aware that if the motive force in many carriage clocks with horizontal escapements be suddenly increased, by pressing the centre wheel for example, the vibrations are seen to quickly become greater, they then fall off, and ultimately are entirely arrested. Can this be due to anything but the accelerating force, first increasing in excess of

the retarding force, then falling short of it, and finally being completely overpowered by this pressure on the cylinder?

319.—Theory and practice have alike shown (**248** to **253**) that as the inclination of the impulse plane is increased from zero, the motive force remaining the same, the accelerating force increases rapidly; but this is only up to a certain limit, beyond which the excessive elevation involves such inconveniences as the following: *too large a cylinder, the employment of too long a power-arm towards the end of the lift, a setting which results from the deficiency of force and involves the application of an increased motive force, etc.*

320.—From what has been said it follows as a necessary consequence, which does not appear to have been hitherto noticed by either authors or practical men, that: *the force acting at the extremity of the power lever* (the radius of the wheel) *is the sole datum from which to ascertain the height of the incline.*

But experience has shown that in watches of average thickness there is sufficient motive force, and that it cannot be increased without at the same time multiplying the sources of wear, and therefore the question now under discussion may be expressed in the following terms:

The motive force being known (assumed to be sufficient) *what inclination must be given to the impulse plane in order to produce an oscillation of about* 270° *without there being any occasion to fear a setting?*

Such a question is beyond the range of calculation, and can only be resolved experimentally. But the results thus obtained will suffice, and the very precise conclusions to which they lead will be found under the heading: *Rules for Determining the Dimensions of a Cylinder Escapement.*

The height of the plane must vary with the size of the watch.

321.—Assuming every detail of a large and small watch to be in proportion, the height of the impulse plane, as compared with the several dimensions of the machine, ought to be greater in the small than in the large watch.

We will presently examine the results of experiment on this subject; but it seems necessary first to study it theoretically.

The angular motion of small balances is more rapid than in the case of larger ones (Theory of the Balance); they move more instantaneously under the influence of the wheel. Hence it becomes necessary to either reduce the number of teeth of the escape-wheel, or increase the height of the incline (**248**).

The question is further complicated in the case of the

small watch; the angular movement of the wheel, or rather the facility with which it commences its motion, appears, when everything is accurately proportioned, to be the converse of that of the balance as regards promptness, and to vary inversely with the size of the watch; this must be considered to be due to the resistances occasioned by friction, and the viscosity of oil, resistances which become appreciable when opposed by forces so minute as those influencing the escapement in a small watch. Moreover, an increased rapidity in the movement of the escape-wheel would entail the expenditure of a greater force in order to overcome inertia (**123**). We have, then, ground for believing that the promptitude with which the wheel commences its motion is not in proportion to the velocity acquired by the balance, and hence it becomes essential to increase the inclination of the impulse plane.

The weight of the balance, as we have seen, has a certain amount of influence on the differences in the heights of these inclines. The existence of such differences cannot be questioned at the present day, but, to the best of our knowledge, attention has never before been directed to them; it can, however, only be doubted by such as are utterly ignorant of the laws of Mechanics and the empirical rules of practical Horology.

322.—In the Swiss and French factories, where unfortunately theoretical instruction is sadly deficient, or even quite neglected, but where there are at the same time a certain number of skilled and intelligent workmen, these two contradictory facts may be observed; a mean lift of 40° in the cylinder escapement has been generally adopted for the last sixty years, and yet on measuring the angle of the incline in a great number of good Geneva watches, both large and small, one is astonished at the immense differences met with. By what unaccountable instinct are these select workmen, who for the most part have the advantage of observations made by their predecessors, led to consistently make the inclination greater when the wheels are small, than with large wheels? Is it not because experience, that great master in horology, has proved to them that it is necessary?

Question the superior class of watch-examiners, and it will be found that they are often unable to time very small watches until after replacing the escape-wheels by others, in which the impulse planes are much inclined.

Influence of the Weight of the Balance on the Height of the Incline.

323.—*The force that neutralizes the inertia of a body varies with its mass.* (**33**).

If two detached pendulums of equal length, say one yard, be suspended freely and one be provided with a very heavy bob, and the other with a very light one, it will be observed that after setting them in motion the latter will come to rest after a few minutes have elapsed, whereas the former will continue its oscillatory movement for a long period.

If, instead of two pendulums, we employ two annular balances without springs or banking pins, but with pivots of equal dimensions, in fact alike in every respect except that they differ materially as regards the weight of rim, it will be found on giving each a movement of rotation that the lighter balance will come to rest in a few seconds, whereas the heavier will continue its motion for several minutes.

An analogous effect will be produced when each is provided with a suitable balance-spring.

324.—All these effects, dependent as they are on the law of inertia given above, a law which we have endeavoured to bring home to the reader by well known examples, clearly show that:

325.—With a light balance, having but little *mass*, such as is met with in very small watches, it is necessary that: (1) *the impulse be communicated more frequently;* (2) *the accelerating force be proportionately greater;* (3) *the friction on the locking surfaces be reduced.*

326.—And that the balances of large watches, having an appreciable *mass*, require as compared with smaller balances: (1) *a relatively greater effort to set them in motion;* (2) *a relatively less force to maintain this motion;* (3) *less frequent application of this impulse*, since the heavy balance continues moving for a longer period.

327.—An increased inclination of the impulse plane in the smaller escapements will satisfy the conditions (2) and (3) of article **325** since the *accelerating* force will be thus increased. As regards (1) every watchmaker knows that it is impossible to regulate small watches when the number of vibrations per hour is no greater than is adopted in the case of large watches.

328.—A decreased inclination in the case of escapements of larger dimensions will meet the requirements (1) and (2) of article **326**, for it will facilitate the starting of the escape-wheel, and at the same time transmit a smaller amount of force. As to (3) it is well known that the balance of a small watch should beat about 19,000 vibrations, 18,000 will suffice for a medium size watch, and 17,000, or even only 16,000, for those of the larger size (see Theory of the Balance).

329.—It is in consequence of the same law, namely, that the greater the *mass* of a balance the greater will be the force required to set it in motion, but when once started it possesses in a proportionately increased degree the power of continuing its movement, that the starting point of the escapement is not identical with the two classes of watches. Thus, *except when the motive force is insufficient for the escapement, small watches should commence going with the second turn of the key, watches of the ordinary size at the third turn, and the larger sized watches, probably not till the fourth turn.* This rule must be understood to be merely empirical, and in no sense binding.

330.—It will be doubtless understood that the mechanical law we are now discussing suffices to explain the well-known fact that in small watches, even though all the parts be made in proportion to those of larger size, the angle of oscillation is always less than in these latter (necessarily causing an increase in the number of vibrations per hour); the balance has less mass, and therefore only a small amount of force is opposed to the resistance caused by the air, oil, friction, etc.

331.—Having proved both theoretically and practically that the inclination of the impulse plane should vary in a certain definite manner with the size of the escapement, it remains for us to determine by careful observation and numerous experiments what heights are most appropriate for each size of watch. Such observations and experiments, considered in conjunction with the principles already laid down as well as those contained in articles **323** to **330**, have led us to adopt the figures given in the following table, and we would add that a prolonged practical experience and a careful examination of many escapements have convinced us of the excellence of these proportions.

The angle of the impulse plane should be such as to produce—

In large watches ...	about 20°	of lift on either side.
	„ 40°	„ total real lift.
	„ 35°	„ apparent lift.
In medium size watches	„ 25°	„ lift on either side.
	„ 50°	„ total real lift.
	„ 40°	„ apparent lift.
In small watches ...	„ 30°	„ lift on either side.
	„ 60°	„ total real lift.
	„ 55°	„ apparent lift.*

These figures cannot be regarded as strictly accurate, for, however carefully two similar watches be constructed, one may be certain that the two wheels are not animated by the same power. But since the differences cannot be very great when they are well made, the data given above will form an excellent guide.

Many watchmakers will express surprise on seeing this table; for they little know, not having any accurate means of verification, and not clearly distinguishing between the two kinds of lift (**306** and **307**), that they often have escapements in hand where the difference between the heights of the inclines is still greater than that indicated above.

332.—These proportions, while being well suited to watches that are thoroughly well made and of a good average thickness, must not, as we have already observed, be regarded as fixed and invariable, but should be somewhat elastic, that is, they should depend on the force at our disposal (**320**); *as a general rule, it is preferable to have the inclination a little too great rather than too small.* It is important to bear this fact in mind not only in connection with the repair of watches, but in its bearing on those in which the cylinder is formed of drawn steel, and only too often badly polished (not to mention the escape-wheel teeth which at times are not polished at all). The friction on the locking surfaces is thus somewhat harsher than it should be, and it becomes imperative, in order to secure fairly good going, to slightly increase the height of the impulse planes, thus augmenting the accelerating force. By so doing we only counteract a great fault by a smaller one; but experience has

* It will be well here to make a practical observation:

Whenever an escapement, while satisfying the conditions given in the above table as well as those with regard to the opening given farther on, performs a vibration which is of insufficient extent, it indicates that the motive force transmitted to the cylinder is deficient; this may be due to bad depths, an inferior mainspring, the watch being too thin or the last mobiles too large, etc.

long since proved the necessity of it in factories where inferior watches or even those of an average quality are produced. This explains the fact which has been noticed by every practical man, namely, that it is possible to judge of the relative qualities of the watches made in different factories by examining the average height of the impulse planes. Thus on inspecting a considerable number of the watches manufactured at each centre of industry we observe that, as a rule, the incline at Geneva is less than at Locle; at Locle it is less than at Chaux-de-Fonds, etc.

In conclusion we would add, that no watchmaker who has thoroughly mastered the principles we have explained, need ever feel at a loss whatever escapement he may be called upon to make or to repair, and that he will always be able, from his own personal observations, to ascertain the exact point of equilibrium between the accelerating and regarding forces.

The opening of the Cylinder and the Form of its Edges.

333.—The several authorities are, as we have already seen, very much divided on the first of these points, and they make the dimensions of the half-shell to vary from 185° to more than 200°, or by no less than 15° or 17°. In practice the half-shell has for a long time past been made rather less than 200°. But since experience has shown that a slight variation on either side of this amount has no appreciable influence on the extent of the vibrations, we are forced to admit that practice has come nearest to the truth, and has had the better of a certain kind of science, but, be it understood, this is improperly applied science.

What have the advocates of a considerable opening in view in their over-anxiety about friction? To diminish the period of rest. What experiments have they made on the subject? What evidence is there that a slight increase in the period of rest has any detrimental effect on the extent of the vibrations, on the wear of the several parts, or on the constancy of the timing? As to a proof, they always forget to give it; and this notwithstanding its being an indisputable fact that escapements with frictional rest go very badly when the periods of rest are much reduced; while, besides this negative effect, we often meet with such escapements, in which

the periods of rest are considerable, not causing any practical inconvenience.

It should be noted that Moinet, in referring to a long period of rest, speaks of the "obvious compensation that may result from it dependent on the amount of this friction."

334.—This disagreement between the several authorities is as easily accounted for as that which occurs amongst practical men.

The question may be looked at from two aspects; two dangers have to be avoided, and one or the other appears to be the more important, according to the point of view from which the question is considered.

When the cylinder is much cut away, or, to use the workshop term, very *open*, the vibration of the balance is somewhat short.

With a cylinder that is much *closed*, the friction during the rest is more prolonged, and the escapement becomes sluggish all the sooner on account of the clogging of the oil.

The two extremes are equally inconvenient, and it therefore becomes evident that by taking a mean we shall secure the best proportions.

But where is this mean?

It cannot be accurately determined, depending as it does on the resistance occasioned by the pressure on the locking surfaces; when this friction is at all harsh, its period should be reduced, and conversely when such is not the case.

In modern watches the half-shell is about 200°, and if the depths are good, and the motive force sufficient, the arc of oscillation will not be less than 265°. With a balance of the usual dimensions, such an amplitude is necessary in order to ensure the watch maintaining its rate.

Take such a watch, and reduce the half-shell to only 185°, all other conditions remaining as before.

Since the actual impulse communicated to the cylinder is represented by the height of the inclined plane, it will remain the same as before. It requires no demonstration to show that, although the impulses are equal with these two cylinders, the supplementary arcs cannot fail to terminate 6° or 8° sooner on either side; this amounts to a reduction in the total vibration, when the cylinder is opened, of about 15°, a fraction of the entire arc which we certainly cannot afford to despise (**343.**)

It must not be forgotten that there is one consideration that overrides all others in this matter: it is the necessity, at all costs, of maintaining the vibrations of sufficient extent, since that alone is a guarantee of the timing, as it renders the balance capable of overcoming the resistances opposed to it. This fact seems to be ignored by the advocates of very open cylinders. It is strange that they should not have noticed the possibility of reconciling these two systems, as we proceed to show: and this result proves once more how advantageous it is to study, especially in the case of an escapement, concurrently on paper and in the watch itself.

The solution of the problem of the opening rests mainly on the form given to the lips. It has been partially solved in practice for a long time past; but this has been either forgotten or disregarded by writers who had not had sufficient practical experience of the mechanism of which they wrote.

The form of the Lips.

335.—Several considerations should guide the watchmaker in determining the best form to be given to the cylinder edges.

It ought to possess the following advantages:—to reduce the wear of the lips by distributing the friction; to render the vibration more free by diminishing, as much as possible, the drop on to the locking surface; to avoid in part the slight delay due to the inertia of the wheel and the oil.

Any change in the opening of the cylinder renders necessary an alteration of the form of the lips.

Engaging Lip.

336.—The best form for the great lip of a cylinder in which the half-shell measures 196° is a very flat curve, as shown at *n*, fig. 4, plate I., and this has been studied experimentally on a large scale. It approximates very closely to a straight line directed towards a point below the incline of the small lip. The outer angle is rounded considerably, whereby the tooth is released more rapidly, and the shock of its entrance is reduced. The internal angle should be very slightly rounded, and the drop may thus be prevented from exceeding what is absolutely necessary.

With such a form the friction of the impulse plane will be distributed as uniformly as possible over the entire surface of the lip; and it is all the more important to take this precaution

against wear since the friction is that known as engaging friction.

An inspection of the figure will show that the rounding off occupies at least 4° of circumference.

337.—The second dotted form *m*, which, except as regards the rounding of its edges and a slight curvature of its surface, is very nearly as left by the file employed in cutting the cylinder, appears at first sight to approximate to the form *n*. It nevertheless differs considerably, indeed to such an extent that in this case the entire friction of the impulse plane occurs against a narrow portion of the extremity of the edge, and the wear will therefore be proportionately greater.

338.—As to the circular form *r*, it is just the one of all others to be avoided, but it is to be found in the great majority

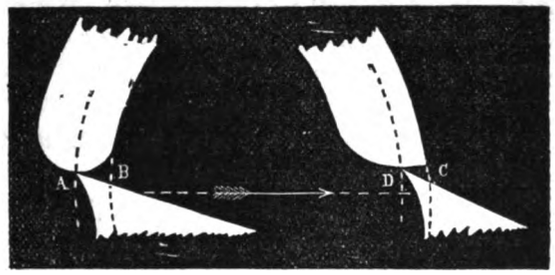

Fig. 27.

of cylinders. All the friction occurs near the middle point of the shell's thickness, so that the drop is increased by the entire space included between the dotted lines A and B (fig. 27).

Disengaging Lip.

339.—The form which the small lip is caused by wear to assume has occasioned surprise to watchmakers who are often called upon to replace old cylinders. This wear, which leaves the external point of the lip untouched, is always greatest at a point which is not touched, or only slightly, by the impulse plane of the tooth, a fact that may easily be demonstrated by means of a somewhat enlarged model. Such a fact proves that the wear is mainly caused by the point of the tooth being badly rounded and insufficiently polished, and by the small drop of the wheel as it leaves the locking surface. Having ascertained this fact it remains for us to explain it.

When the incline of the disengaging lip is of insufficient extent, the friction of the tooth occurs solely against the middle of this incline, and the heel of the tooth escapes from the lip

without ever coming in contact with its external portion; the drop will thus be increased by the space between the lines C and D, fig. 27.

The form of lip that is best adapted to avoid these failings is a long flattened curve, so arranged that the tangent drawn at its centre (A *p*, fig. 4, plate I.) passes through the middle of the half-shell. We thus obtain the curved face *d i b*.

If now this curve be compared with any other less elevated, for example, *d v* in the same figure, it will be seen that the space traversed before an impact against the lip occurs will be reduced in the first case by the entire distance included between the curves *d i b* and *d v*. The friction of the tooth will no longer take place at the middle of the lip, but at its external point, and by this means the drop will be reduced and the impelling force increased, since it will be applied at the extremity of a longer resistance lever. With regard to this friction on a single point, there is no occasion to anticipate any ill effects from it, for it is of the nature of *disengaging* friction, and there is hardly a case on record of wear commencing at the external corner of the exit lip; it is always at the internal angle of its incline that this lip is first affected.

In a well proportioned cylinder, this incline would occupy about 10° of circumferential arc, as is indicated in figure 4, plate I. Compared with the dimensions given by other authors, this amount may appear considerable, but if the dimensions of the disengaging lip in several cylinders by good makers be examined, one is surprised to see how closely, if not indeed exactly, they conform to it.

It will thus be seen that the form of the lips may have a very great influence on the timing of the watch.

The Extent of Opening.

340.—Having now ascertained the best form to be given to the lips, it will be easy to fix upon the exact opening of the cylinder. To the half-circumference of 180° add 2° additional, for experience has shown such an amount to be necessary, in order to ensure certainty in the performance of the several functions. We thus have a total of 182°; add to this amount 4° on account of the rounding of the great lip (**336**) and 10° for the incline of the small lip (**339**), and we finally obtain 196° as a measure of the half-shell.*

* If it be remembered that the rounding of the points of the teeth must always cause the point of contact to be slightly below the line joining the point and heel (or the diameter of the cylinder), that it is impossible for all these touching points or for

This amount of opening combines the advantages aimed at by the advocates of both systems; the impulse is applied to the cylinder under very advantageous conditions, and, since from the form of its lips all the good qualities of a more closed cylinder are secured unattended by its disadvantages, the extent of the vibration so far from being limited is always augmented. The friction that occurs on the locking surfaces differs very little in amount from that of a 185° cylinder, and the more effective impulse combined with a shorter drop gives an amplitude of oscillation which varies very little from that obtained with a 200° cylinder under ordinary conditions, and may even equal or exceed this amount.

Experience leaves no room for doubting the advantage of using these proportions.

On examining the forms of the cylinders produced by the best makers of the old school, who were taught by long-continued observation, handed down from father to son, we find them to be in accordance with our principles. The amount of opening which these skilful watchmakers caused to be adopted as most satisfactory in the factories of Geneva, gave a half-shell of very little less than *seven-twelfths* the external diameter of the cylinder, proportions that coincide accurately with those given above.

341.—The opening should be the same for every escapement, with a lift of 40° or more, but it must be slightly augmented for the larger class of escapements with a lift of not more than 30°, in order to prevent their setting. We have already learnt enough to know that in such a case no change of the kind could have any ill effect.

342.—The figure 196° as a measure of the half-shell answers the requirements of practice in the very great majority of cases; but it is not, neither can it be, in any way absolute, since the friction on the locking surfaces should depend on the greater or less energy of the impulse.

the heels to lie on the same circumference, or for the impulse planes to be of mathematically equal length; if we take account of the play of pivots in their holes, the mechanical difficulty involved in the very delicate operation of rounding the great lip to the requisite extent; and, lastly, when it is observed that, as the locking point of the tooth is at *s* (fig. 4, plate I.) the interval *s d* in a cylinder barely one-hundredth the size of the figure will be entremely minute; it must be admitted that no less amount can be relied upon to ensure the uniform working of the escapement.

The locking point *s* is in practice about 4° from the edge of the lip.

When the nature of this friction, or the too great length of its period, renders the escapement sensible to differences in the consistency of the oil, it is always best, instead of attempting to diminish the period of rest by any means, to reduce the diameter of the cylinder, that is to replace it by a smaller one. Such a change of course necessitates a new wheel as well, but this need occasion no difficulty at the present day, since wheels and cylinders of all dimensions can be obtained of the watch material dealers.

EXPERIMENTS.

343.—The following experiments were all made with a single cylinder acted on successively by three wheels, in which the height of the impulse planes varied.

The opening was twice increased, the edges being re-formed and re-polished so that the half-shell approximately corresponded in the three cases to 196°, 192°, and 185°.

Observations made on the going in each case lead to the conclusion that in the generality of modern watches:

1. There is a loss in the arc of vibration of at least as great an angle as that by which the half-shell is reduced, and sometimes even more.

2. A change in the opening has less influence on the extent of vibration than a change in the inclination of the impulse planes; and indeed this must be the case, for the force exerted by the plane remains very nearly the same, even although the opening be somewhat increased or diminished.

3. In the great majority of cases (the exceptions being certain watches with large heavy balances and a slow period of vibration) if the opening be made greater the impulse planes must have an increased height.

4. By opening the cylinder so that the half-shell only exceeds a semi-circumference by the projection of the lips (giving a half-shell of 185°), and depressing the incline until it acts tangentially, a considerable proportion of the vibration is sacrificed in utter waste.

Concluding Observations on the Opening.

344.—It is a fact well known to watch-jobbers that with cylinders open nearly to the centre, such, for example as those of 185°, the vibration is insufficient and can never be regulated satisfactorily. The edges of such cylinders should be very little inclined or rounded; but, since in practice the precise

amount indicated by theory is almost always exceeded and the 2° allowed as security is thus reduced to nothing, it results that the tooth falls against the corner of the lip and even on the lip itself, for the point of this tooth is not a sharp angle, and the point of contact is always slightly below the line of the straight incline. Then again one can never be certain that the middle of this plane coincides with the centre of the cylinder, that the wheel is truly centred on its pivots; and even should this condition be satisfied and all the points of the teeth be on one and the same circumference (the cutting, hardening, and tempering are quite sufficient to distort some of them), it is always found to be necessary to bring the cylinder nearer to the escape-wheel, and thus give rise to excessive drop on the outside or a constrained action within the cylinder. It must be noted also that the friction on the small lip is borne by an angle and not a surface, so that the wear is rendered very rapid; that the drop on to the locking surface, owing to the form of this lip, is very severe, as we have already shown (**339**); and we cannot fail to see that the vaunted improvements resolve themselves into the following: greater difficulty in the adjustment of the escapement, more rapid wear, shorter vibrations, greater trouble in timing, etc.; and that such improvements in horological art may as well be left undone.

We have here another example of the danger of purely graphical or theoretical solutions or even of those deduced from experiments with large apparatus. Their proportions, which cannot be the same as on the small scale, give a false idea of the necessary spaces or *plays* and the increased weights of the mobiles entirely alter the ratios of the forces, velocities, etc.

In order to show the importance of this observation, it will suffice to quote this geometrical proposition: The weights of similar bodies of the same substance are to each other as the cubes of their homologous sides (**137**).

Size of the Axes in the Escapement.

345.—In axes of the escapement we include not only that of the cylinder but also the escape-wheel axes.

When the cylinder is too large,* a very common fault in small watches, there cannot be a balance between the motive power, the size of the balance and the radii of rest; all the

* It must be understood that when speaking of cylinders and pinions that are too large or too small, or of wheels that are too large, we do not employ these terms

friction is increased and the retarding force becomes predominant, etc. Thus the vibrations become sluggish, or, as it is expressed in the trade, there is a falling off in the crossings, and as soon as the oil thickens the regulating of the watch becomes difficult.

With a too small cylinder, generally in consequence of the wheel having too many teeth, the impulse communicated is slight, since the length of the arm by which it is transmitted is excessive; the impulse plane is too short, and passes the lip too rapidly, so that it becomes necessary to increase its inclination: these two circumstances render the employment of an increased motive force essential, and every increase in this force above a certain defined limit is a source of wear and irregularity.

Should the escape-wheel pinion be too large in comparison with the radius of the wheel itself, or if it should be low numbered, any faults and inequalities in the depth will have all the more effect on the escapement. With such a pinion the oscillations of the balance will not be performed with the requisite freedom, but will be, as it were, constrained.

Size of the Escape-Wheel.

346.—It is a mistake to make the fourth wheel of the train (the seconds wheel) very large, especially if, as is generally the case, it be with a view to employ a large escape-wheel. The size of the wheels ought to decrease as the force to which they are subjected diminishes.

The fourth wheel should be high rather than low numbered, and the teeth should be somewhat fine; by this means a pinion can be employed of the size best suited to the escape-wheel. As regards this latter, if it is essential that it be of considerable diameter, the number of teeth must be increased by one, and the wheel made as light as possible. But it must be remembered that it is usually preferable to have the *escape-wheel somewhat small, and the moderator heavy, rather than a large wheel and a light balance*, but this principle must not be carried to an excess.

in the sense ordinarily understood in the trade. We do not refer to the actual dimensions of the several parts but to their relative sizes as compared with the rest of the mechanism. We are here only discussing the size of the escapement, as a whole, compared with that of the entire train.

Concluding Observation.

347.—In the cylinder escapement, as indeed in all escapements, absolute perfection is impossible, and it must always be regarded as a relative quality. And we must never forget that excellence is to be secured by a general harmonizing of the several parts, by a certain balancing of the forces in action, and, finally, by the adoption of a general basis that is somewhat elastic and will be easily ascertained by experience.

Any watchmaker who will take the trouble to carefully examine, measure, and note down the proportions existing in even a small number of watches that are keeping good time, and whose vibrations are full and free, will very soon be able to judge at a glance whether the thickness of axes and the dimensions of wheels and balances approximate sufficiently to the best size possible; whether, in short, they are such as to ensure the regularity of going of the watch.

We have dwelt at considerable length on the principles involved; for principles alone can beget *intelligent* workmanship, the very converse of blind and helpless routine. We are now in a position to enter on the practical application of these principles, and will proceed to do so by deducing the proportions of an escapement in accordance with the demands both of theory and experience.

CHAPTER IV.

THE CONSTRUCTION OF A STANDARD CYLINDER ESCAPEMENT AND EXPERIMENTAL DATA.

348.—This chapter is especially addressed to manufacturers, who are, as a rule, too much disposed to imitate the forms ordinarily met with. By employing the knowledge that we now trust the reader possesses, he should be in a position to plan a method for himself, should the one which we proceed to propose not give entire satisfaction. But let him be careful to remember this fact; the success of a particular escapement does not depend on so-called *secrets*, which the charlatan pretends that he possesses; success in the exact measurement of time is *always* the result of a judicious application of the laws of Physics and Mechanics, and this fact must never be forgotten.

Natural Compensation.

349.—Natural compensation is the result of an equilibrium, more or less perfect, between the causes which tend to produce gain and those producing a loss. These causes arise, on the one hand, from changes occasioned by time in the nature of the several frictions, and the clogging of the oil; on the other hand, changes in the condition of the oil, and in the expansion or contraction of the balance-spring and balance caused by changes of temperature.

Oil coagulates in the cold, becoming liquid on heating.

When the oil is thus congealed by the cold, the several parts of the escapement are checked in their motion and the vibrations nearly always become shorter and slower. But if these causes of loss are met to a corresponding extent by a contraction of the balance and its spring (cold increases the power of the latter), the watch maintains an approximately uniform rate.

On the oil being liquefied by heat, the balance has more freedom, since the friction, especially that on the locking surfaces, is less, and its oscillations become greater; and, taking into account the expansion of the balance-spring and balance, there are three causes of loss set against one cause of gain; this latter consists in the increased tension of the balance-spring owing to the arcs being of greater extent.

If the increase in the tension of the balance-spring has as great an effect in producing a gain as its diminution, etc., had in causing a loss, the balance will be impelled backwards with all the greater energy, and as the oscillations are, at the same time, longer and more rapid, the conditions will remain the same on the whole, and equilibrium will be maintained.

It thus becomes evident that this equilibrium in a watch of which all the parts *are well proportioned* depends mainly on the length of the balance-spring, the correcting effects being excessive with a very short spring and deficient when it is of considerable length; we see, then, why dead-beat undetached escapements, which require oil, should not be provided with balance-springs as long as those employed for detached escapements (**269** and following articles).

It will be seen that the radius of rest has an important part to play. When, for example, a cylinder is too large, the friction on its locking surface is very considerable as com-

pared with the energy of the balance; and since the causes of variation due to friction, especially as depending on the state of the oil, go beyond the very narrow limits within which the correcting influence of the balance-spring is efficient, equilibrium between the gain and the loss becomes impossible, and the watch can never be properly regulated.

Natural compensation does not fully come into play until the watch has been going some months, for then the oil has acquired a certain consistency, and the changes introduced in it by heat and cold are less marked.

Action of the Balance-spring in Natural Compensation.

350.—With regard to the actual effect of the balance-spring in natural compensation what we have just said should suffice for the present, but we feel compelled to revert to one point.

Some practical men who believe, simply because they have heard it stated so, that the balance-spring works all the better as we increase the number of its coils, whatever be the escapement to which it is to be applied, have been at a loss to explain why, in our first edition of the *Treatise on Escapements* we recommended the use of balance-springs shorter than those in favour at the time.

They would have understood the reason for our recommendation had they read with proper care the simple observation that followed, namely: that when the isochronism is destroyed or masked by causes inherent in the nature of the escapement, an isochronous balance-spring is nothing but an inconvenience on account of its great length.

We would here add that to discuss the question whether short springs are preferable to long ones would be mere waste of time and could result in no good. In horology everything must be relative. Whatever be the escapement under consideration, it requires neither a *long* nor a *short* balance-spring, but one that is suited to its nature and mode of action, that is to say, the length must bear a definite relation to the extent of the arcs of vibration, etc.

In this same first edition we further stated that:

"If we consider a dead-beat escapement to which oil is essential, such, for example, as the horizontal one, it will be observed, as we have already pointed out, that heat renders the oil more fluid, the balance larger, the balance-spring longer and

more pliant, so that an elevation of temperature introduces several causes of a loss in the rate.

"As the length of the balance-spring is increased, its elastic force becomes less; so that of two springs differing materially in length but having the same force, that is producing an equal number of vibrations in equal times at a certain fixed temperature, the longer will suffer most from any change in the degree of heat, and if the watch is timed successively with these two springs at the given temperature, a rise will occasion more loss with the longer than with the shorter balance-spring."

We assumed then, on the strength of information which we considered deserving of confidence, that at high temperatures a short balance-spring is the more effective. At the same time we could not help doubting the fact; for, from a theoretical point of view, the dilatation would be proportional to the dimensions of the spring, and the result therefore should be the same, whichever we employ.

To solve the question experimentally by using watches provided with frictional rest escapements, was a matter of considerable difficulty, owing to the fact that in such escapements the effects produced are very complex, and one could not with certainty isolate that due to the balance-spring.

After consideration we decided to construct a small instrument in which springs of various strengths could be held by one extremity while the free ends were charged with a small weight and the following experiment was made with it.

EXPERIMENT WITH ELASTIC BLADES.

351.—The lengths of the several springs were first so adjusted that their vibrations were synchronous at a temperature of 12° C. (53° F.), and the temperature was then raised to 35° C. (95° F.) A want of accord in the vibrations was at once perceived, and this effect was produced on several times repeating the experiment; but with this important difference, that the successive experiments gave results that were somewhat variable, and at times even contradictory. This was at first sight calculated to perplex the observer.

We are unable to draw any precise conclusions from these experimental results, which we shall after repeating fully describe in the article specially devoted to the study of the balance-spring; yet, by induction from the observed facts, we feel justified in accepting as proved these propositions:

1. Theoretically, that is to say with a blade which is thoroughly uniform and homogeneous throughout its entire length, the effect produced by a change of temperature being always dependent on the dimensions of the object, the exact length of the spring is a matter of indifference (we are here only discussing the resultant effect of changes of temperature, for everyone is aware that a difference in the length of the spring modifies the velocity of motion of the balance);

2. The facts of observation, which appear to run counter to theory, must be attributed to three principal causes: a constrained condition of the molecules and a want of homogeneity in the blades, due to careless rolling, whereby unequal dilatation and irregular displacement of the coils are occasioned, and a quivering, which is proportionately greater with the longer spring subjected to an increase of heat (**1400**).

MEANS OF PRACTICALLY DETERMINING THE DIMENSIONS OF A CYLINDER ESCAPEMENT.

352.—We can adopt as the main datum either the size of escape-wheel, or diameter of the balance.

It is more reasonable to select the latter; but in that case we incur practical difficulties, which are avoided if, as we now proceed to do, the diameter of the wheel is adopted as a starting point or unit of measurement.

We will assume ourselves to be engaged on a watch movement of the calliper generally met with at the present day, but in which the radii of the wheels have been selected in accordance with the decrease in motive force (**266** and **346**), and where the pivots are no larger than is ascertained to be absolutely necessary for their due solidity, and to ensure a constancy in the friction by making the rubbing surfaces of sufficient extent (**41**).

The size of the escape-wheel is, then, determined as well as that of the cylinder which is dependent upon it.

To Ascertain the Height of the Impulse Plane and the Size of the Balance.

353.—Having found from the table on page 169, what angle of lift is best suited for the dimensions of the watch to which the escapement belongs, three wheels must be made with inclines of different heights, such that one gives the exact amount of this lift, the second an angle somewhat less, and the third somewhat greater (see **255** and following articles).

The points and heels of the teeth must be sharply defined, being simply polished with rouge so as not to feel rough to the nail.

354.—The table in article **388** will give the dimensions of a temporary balance as deduced from the diameter of the cylinder.

While referring to the chapter specially devoted to the consideration of the balance for details relating to this subject, we would at once point out that, since the *mean* diameter (**103**) is the only legitimate one, it necessarily follows that we may have numerous balances, of the same size and weight, in which the matter is variously distributed, and they may thus be still too *large* or too *small*, having regard to the number of oscillations performed in a given time with one and the same balance-spring.

We have employed the following distribution of the matter of the balance with advantage.

The total weight having been decided upon in the manner explained in the chapter on Balances, or by a systematized examination of a number of well timed watches, this weight is divided by 12. Then the balance is so made that its weight is thus distributed:

In the three arms (taken together) about	1	twelfth.
„ small central ring . . „	1	„
„ rim . . . . . „	10	twelfths.
Total	12	„

This result can be easily obtained, if a few preliminary experiments are made, first separating the arms from the rim, and then from the central ring.

When the metal is sufficiently firm, the weight of the rim may be considerably greater in comparison with that of the arms and centre, but we are here discussing the brass balances employed in watches of the ordinary size. If the arms of a balance formed of this metal cannot be reduced to the proportion given above without becoming too weak, we may unhesitatingly conclude in almost every case that the matter in the rim is deficient, and that the balance will be found to be too light as a moderator (**1342**).

355.—The balance after being completed is fixed to the cylinder by means of gum-lac, and a balance-spring taken from a well timed escapement of the same size is temporarily fixed to it; the first experiment may then be proceeded with.

We say temporarily because it may very probably be found that either the spring or the balance or both are not quite rightly proportioned.

EXPERIMENT.

356.—The three wheels are successively caused to act on the cylinder, and that one is selected which gives a total balance oscillation of 270° and a minimum lifting arc. The incline in this case is such as to produce the greatest amount of useful effect (**224**).

We know then:

The diameter of the escape-wheel.

The diameter of the cylinder.

The height of the impulse plane.

It only remains to ascertain and verify the *moderating* power of the balance, and the *regulating* power of the balance-spring when employed in connection with it.

Half-Timing.

357.—A good moderator owes its qualities to a judicious adjustment of its weight in relation to the extent of oscillation. In other words, its mass must be so great that it is not at the mercy of the motive force and, moreover, its velocity must be sufficient to prevent its being influenced by changes in the density of the air, variations in the friction, and the shocks that occur in wear (Theory of the Balance).

This condition having been satisfied, the spring and balance are removed and replaced by a small pendulum of equal weight, attached to the cylinder in a similar manner.

The watch is placed at a convenient inclination (see the *Twelfth Experiment*, page 141) and set in motion by forces of different intensities. By varying the length of this pendulum, the bob of which can be fixed in any position by means of a clamping screw, the point is ascertained that renders it least sensitive to variations in the impelling force.

Having determined this, the movement must be placed in the oven and the adjustment of the pendulum further continued until there is a gain of about two minutes in passing from 37° C. (99° F.) to 0° C. (32° F.).*

The small pendulum thus obtained will enable us to deter-

* When designing a *standard* escapement, regard must be had to the precise amount of the cylinder opening (**334**). If two cylinders be of the same diameter and similarly circumstanced, but one more closed than the other, it will fall off in its rate somewhat sooner.

mine the dimensions of the balance (by means of the table at the end of the work on *The Relation of the Pendulum to the Annular Balance*).

In the absence of such a table one might employ the method indicated in the eleventh experiment, page 140.

358.—The intensity of the motive force can be caused to vary by giving only a half-turn to the mainspring in the first instance, and four turns subsequently. It would, however, be much better to adjust on the central axis a kind of screw arbor, carrying a light pulley of sufficient size. By means of a silk thread coiled on this pulley, and small weights, varying forces could be brought to bear on the system, corresponding to different degrees of winding up of the mainspring. This mode of experimenting has the advantage of enabling us to keep the mechanism several hours in action under the influence of any predetermined force, and we are the better able to observe the differences in timing occasioned by a diminution in the arc of vibration. It is as well in such experiments to employ oil that is somewhat thick.

To ascertain the most advantageous Relation between the Balance and its Spring.

359.—We now possess the main dimensions of the escapement. But the most difficult portion remains to be done; for, although we may be satisfied that the several elements are good; that the lifting arc will utilize as much of the motive force as possible; and that the sensibility of the moderator to variations in this force is reduced to a minimum, we still know that the moderator is affected by every alteration due to changes of temperature — changes which are very marked from summer to winter—and that it is further dependent on the action of the balance-spring.

The balance-spring borrowed from a movement of the same general dimensions will nearly always occasion an increase in the rate during the long arcs due to an increase in the motive force (**265**).

But cylinder watches of modern make have nearly always a tendency to lose as the extent of the arcs of vibration gradually becomes less; this reduction is brought about by a thickening of the oil. It will be manifest that, if the effect of the balance-spring becomes weaker as this amplitude is diminished, the losing on the rate will be all the more marked.

360.—We conclude from what precedes that, in an escapement constructed on theoretical principles (**269** and **270**), the irregularities due to temperature and to the retarding action of the oil may, for the requirements of every-day life, be counteracted by the balance-spring, but not by it alone. It may probably be necessary to alter the weight of the balance.

What then is the precise effect of the balance-spring? It acts the part of a connecting link between the motive force applied at the extremity of the radius of the wheel, and the energy possessed by the mass of the moderator in virtue of its motion, an energy which must be controlled by this spring.

It thus has a very difficult part to perform, for it is required to render uniform the movements of the moderator in opposition to the variations of the prime mover, or very nearly so. Now this regulating property of the balance-spring is confined within very narrow limits, and if the resistance occasioned by these two opposing forces becomes excessive, the regulating power of the spring may be entirely destroyed or rendered nugatory; instead of regulating the action, either of the balance or the motive force, it will thus be entirely under their control.

361.—There are, then, three forces, represented by the prime mover, the weight of balance, and the elasticity of the balance-spring; and they must coexist with a certain determinate balance of power on the maintenance of which depends whatever regularity is secured in the movements of the moderator.

But of these three forces in the case we are considering, the motive force is fixed or very approximately so, for it has been settled on the basis of the watch whose rate is satisfactory, and in which the impelling force could not be materially increased without inconvenience.

We possess also a balance of dimensions that afford some hope of securing good timing.

Nothing remains, therefore, but to procure a balance-spring that is well adapted to the balance, remembering that the weight of this latter and its spring are correlated or, at any rate, mutually dependent (**262**).

We have thus arrived at the final and decisive operation connected with the timing, the real test of the value of a time-keeper.

If the workmanship be of the best, the materials good, and selected with judgment, any variations that occur in the rate of

a watch on changing its position are due to a change in the total amount of friction.

Between summer and winter, between the temperature of the watch pocket and 0° C. (32° F.), or a little above this latter point, the temperature of the watch varies through a range of about 40° C. (72° F.) The loss that occurs in the winter is mainly occasioned by the increased resistance experienced by the surfaces of contact, when rubbing against one another, through the thickening of oil, and doubtless also from a change in the molecular arrangement of these surfaces themselves.

The main object of timing then should be to equalize, by means of the changes of temperature and alterations of position of the machine, the resistances arising out of friction.

When this equality has been approximately secured, we may proceed to select the most suitable balance-spring (see *Springing and Timing*).

The (new) watch having been regulated by means of the index for its long arcs, so as to gain about 2 minutes in every 24 hours, should, as we have already pointed out, gain about the same amount with a change of temperature from that of the pocket to 0° C. (32° F.); this gives a total amount of about 4 minutes, and the length and form of the balance-spring should be such as to give an additional acceleration of 5 or 6 seconds in 12 hours, on account of the small arcs, that is to say when the last coils of the mainspring are in action.

As the dimensions of the escapement are diminished, the acceleration ought to be more pronounced. In small size watches it sometimes amounts to as much as 10 or even 15 seconds.

362.—It will be understood that this double acceleration, effected partly by the balance-spring and partly by the general arrangement of the several parts, is intended to counteract the loss in the rate which is occasioned by the thickening of oil and increased friction. The causes of gain when the temperature is depressed are, as we have explained, the contraction of the balance and its spring, and the increased rigidity of this latter; this triple effect is constant, always occurring to the same extent, whereas the loss occasioned by friction and decreasing strength of the mainspring not only at times exceeds the gain due to other causes, but is continually becoming more and more marked. It is essential, then, that the causes of loss be rather more than neutralized.

After going for a few weeks the watch will be found to be correctly timed for the temperature of the pocket (about 37° C., or 99° F.), and on cooling it to 0° C. (32° F.) it will gain several minutes in 24 hours, more or less according to the size of watch.

363.—Providing the force is transmitted by means of a good train and sound depths, so that all sudden jerks in the impulse are carefully avoided (for they destroy the sort of equilibrium that characterizes an efficient escapement), and if the oil is not of an inferior quality, the watch will itself maintain the rate, and will go satisfactorily for a much longer period than the few months, at the end of which we are now-a-days compelled to move the indices of cylinder watches.

RECAPITULATION.

364.—We will conclude with what amounts to a summary of the above considerations.

When the value of either one of the three main elements:

The motive force;

The mass of the balance; or

The elastic force of the balance-spring;

experiences any abnormal change (which may be brought about by variations in the friction, a badly-shaped balance-spring, or one that has lost its elasticity, etc.), the general accord between the parts is destroyed and the timing must be done over again.

To re-establish this accord it will clearly be necessary to repeat the operations already described, to vary the value of each element, carefully noting the results obtained during this process of timing.

The reader will, we trust, fully understand this subject when he has, in addition to the above, mastered the chapter on *Springing and Timing* and, prior to that, the following account of certain experiments of F. Berthoud and L. Tavernier:

NOTE ON SOME OBSERVATIONS OF BERTHOUD, JODIN, AND TAVERNIER.

365.—Jodin's experiments convinced him that a balance which is too large as compared with a given cylinder escapement occasions a loss with any increase in the motive force, and the converse is the case when the balance is too small. Fétil justly observes that this observation should have led him to ascertain the best proportion for the radius of rest of the cylinder to bear to the radius of the balance. Jodin, however, like F. Berthoud, appears to have taken no notice of this fact.

Louis Tavernier* often observed the following phenomenon in the horizontal watches of his day: "If the friction of the balance is excessive, that is to say if the diameter of the cylinder is too great as compared with that of the balance, the rate of the watch will appreciably increase as the oil becomes more and more thick."

The watchmakers of that period, guided solely by experience, constructed the escapement so that the diameter of the cylinder was to that of the balance as 1 is to 14.

Such a proportion is unsuited to the great majority of the escapements of the present day. This fact is chiefly due to the suppression of the fusee, changes in the radius of the wheel and the number of its teeth (many of the large escape-wheels employed in old watches had only twelve or thirteen teeth), and lastly, to the substitution of steel for brass in the construction of these wheels.

EXPERIMENT OF F. BERTHOUD.

366.—"A horizontal watch, beating 18,720 vibrations per hour, gained 4 minutes in every 24 hours when at the temperature of the pocket.

"When suspended and maintained at a temperature of 4° C. (39° F.) it lost 14 minutes 30 seconds in 24 hours as compared with its rate at the more elevated temperature.

"But the arcs described by the balance were very great, the balance itself was very small and had large pivots, so I replaced it by a new one that was considerably heavier; when the watch was regulated there was great risk of a setting.

"When passing from the higher to the lower degree of heat, instead of losing 14 minutes 30 seconds in 24 hours, the watch only lost 3 minutes in the same period.

"Still, not being content with such an approximation, I diminished the balance pivots in order to reduce the friction, and again regulated the watch at the upper temperature. It

* Louis Tavernier lived at the commencement of the present century, and was one of the best watchmakers in Paris. He studied the cylinder escapement with great care and with considerable success, but unfortunately the greater part of his works appear to have been lost. After a prolonged search we have only been able to discover traces of them.

Circumstances of this kind are but too common in Horology, in which numberless experiments have been attempted. For want of publishing, or through the carelessness of authors, their results have too often remained unknown or been appropriated by plagiarists.

now lost 2 minutes in 24 hours, whereas, in its initial condition, the application of heat had caused a gain.

"In the condition to which it had now been brought it gained 11 seconds in 13 hours if hung by the bow, and, if laid flat, 55 seconds during the same period; that is to say, 44 seconds more than when suspended.

"Lastly, on somewhat reducing the weight of the balance, the rate when cold only exceeded that when hot by 20 seconds in 24 hours."

367.—"It follows," says Berthoud, "from this experiment that:—

"1. On increasing the weight of a balance (by which its true or mean diameter must be increased), the extent of the arcs of vibration is reduced, and the loss from hot to cold, from being 14 minutes 30 seconds, becomes 3 minutes.

"2. On diminishing the pivots the friction is made less, and the loss of 3 minutes with the lower temperature is changed into a gain of 2 minutes.

"3. Lastly, on somewhat reducing the weight of the balance the arc of vibration is increased to such an extent that the watch does not vary with a change of temperature."

SECOND EXPERIMENT OF BERTHOUD.

368.—"A horizontal watch had a balance weighing 4·5 (old French) grains (0·24 grammes); it was far too light as compared with the motive force, 7·5 drachms (28·6 grammes). The watch in this condition when timed for the pocket lost 10 minutes in 14 hours if exposed to a temperature of 1° C. (34° F.)

"After diminishing the force of the mainspring till it was only about 5·25 drachms (about 20 grammes), and thus rendered the watch very liable to set, I timed it similarly for 37° C. (99° F.), and on keeping it cold for 14 hours it still lost 5 minutes." Only half the error then was neutralized.

Berthoud, considering the resistance (which he attributed solely to the oil) to be too great as compared with the force exerted by the balance, had a spring made to carry 8·25 drachms (31·5 grammes), and replaced the balance by one weighing 7 grains (·37 gramme). The watch having been again timed for the pocket, and exposed to cold (1° C., or 34° F.), gained 2 minutes in the same period of 14 hours.

"Hence," remarks Berthoud, "it will only be necessary to

reduce the weight of the balance in order that this watch may not gain either when it is hot or cold."

369.—Berthoud concludes, as the result of his experiments, that a new watch should be so adjusted that it gains on being heated. This gain, resulting from the greater freedom of the mobiles and the excess of motive force, will correct itself when the arcs of vibration diminish in extent, that is to say, when the rubbing surfaces have come to fit each other by their mutual friction, when the oil has attained that consistency which continues for a long period almost unchanged, and lastly, when the mainspring has lost the overplus of force that a new spring is always found to possess.

THIRD EXPERIMENT OF BERTHOUD.

370.—"I made a watch giving 7,200 vibrations per hour. The balance weighed 19·75 grains (1·05 grammes), the lifting arc was 45°, and the fusee made a turn in $4\frac{1}{2}$ hours.

"I had a spring made that balanced a weight of 3·5 drachms (13·4 grammes) at a distance of 4 inches from the centre of the fusee; but the arcs described by the balance were excessive, so that the watch gained when heated. I weakened the spring till it only carried 3 drachms (11·4 grammes), and after timing the watch the balance described arcs of 240°. When thus adjusted the watch was regulated, and neither changes in its position nor in its temperature could disarrange its timing, while formerly heat had made it gain considerably; as a matter of fact, heat made it lose 1 minute in 24 hours; by slightly reducing the weight of the balance I was enabled to adjust it so that change of temperature had no effect whatever. This experiment proves that an increase of $\frac{1}{2}$ a drachm (about 2 grammes) in the motive force has a very marked effect, and that it is of the first importance to proportion the motive power to the regulator by which it is controlled."*

EXPERIMENT OF TAVERNIER.

371.—Louis Tavernier (**365**) noticed that when the cylinder is too large, that is when its diameter is too great as com-

* Carefully conducted experiments are always instructive, and contain some definite information. We are, therefore, very willing to describe these results of Berthoud's, although we feel compelled, especially in the matter of the cylinder escapement, to reject the greater part of the observations with which he accompanies them. Berthoud seems to have performed his experiments with care, but the argument, which leads up to and explains them, is too often specious, and dictated by a manifest desire to run counter to his contemporaries.

pared with that of the balance, the watch will gain as the oil thickens.

He states that, "in order to avoid this inconvenience, the dimensions of the cylinder must be so far reduced that the watch, when new or freshly cleaned, gains from 3 to 4 minutes in 24 hours, with a change of temperature from that of the pocket to the freezing point of water."

The reasons which he assigns for such a practice are:

1. In undetached escapements the thickening of oil at the several pivots of the train has but little effect (providing these pivots are not larger than is necessary);

2. The contraction due to cold will reduce the diameter of the balance and increase the force of its spring, and these two causes tend to produce a losing rate of the watch (this is doubtless a clerical error, since a *gain* will result), and the thickening of oil on the resting surfaces of the cylinder will cause it to lose; but it is not essential that this loss due to the oil be exactly counteracted by the gain resulting from contraction, for this latter effect is always reproduced to the same extent, whereas the friction continually increases as the oil becomes thicker.

It will be evident, from the above quotations, that neither Berthoud nor Tavernier seems to have suspected the length and form of the balance-spring to constitute an important element in the timing.

Conclusion.

372.—The careful study of the theory of escapements with frictional rest, and the exhaustive practical details which precede, as well as those that follow, are sufficient to enable any intelligent watchmaker to design a *standard* escapement, that is an escapement correctly proportioned for a watch of any given size; it will then be only necessary to copy this watch with as much accuracy as is possible. One can then feel confident, after submitting each copy to a few simple tests, that it is a first-class machine, capable of keeping time with all the accuracy that can reasonably be expected from an escapement in which the rest is frictional.

EXPERIMENTAL DATA

BEARING ON THE DIAMETERS OF THE BALANCE AND CYLINDER.

We shall here supplement the data derived from our own experiments by others, all of which rest on the authority of men who are both skilful horologists and trustworthy observers.

373.—GANNERY.—Experiment shows that the diameter of a balance, such as that met with in ordinary Swiss watches, should be to that of the cylinder as 182 : 11 :

Or as 16·5 : 1.

Hence, when the diameter of cylinder is 0·8 millimetres (0·0315 ins.), the diameter of balance will be 16·5 × 0·8 or 13·2 millimetres (0·52 ins.)

374.—M. RODANET obtained excellent results by employing the following proportions, which he submitted to a prolonged investigation:

40 mm. (18 line) Watch—Balance : Cyl. : : 18 : 1.
42 mm. (19 ,,) ,, B. : C. : : 17·3 : 1.

375.—F. BERTHOUD.—A fusee watch performing 7,200 vibrations per hour. The escape-wheel, with 15 teeth, had a diameter of 7·8 millimetres (0·307 ins.).

The extent of a complete oscillation was 280°. Diameter of balance, 21·4 millimetres (0·843 ins.). Ratio to that of the cylinder:

B. : C. : : 20·7 : 1.

Berthoud, in his writings, recommends large cylinders and very short arcs of vibration; but, as will be seen from this example, necessity sometimes compelled him in practice to go counter to his own principles.

376.—M. PIERRET has made a prolonged theoretical and experimental study of this escapement, and constructed watches of the following dimensions:

Size of plate, 32 to 34 mm. (14 or 15 lines).

Escape-wheel, 13 teeth; diameter about a quarter less than usual.

Diameter of balance, 15 mm. (0·59 ins.).

First ratio, B. : C. : : 18 : 1.
Second ,, B. : C. : : 20 : 1.

The balance is, in these watches, somewhat heavier than is customary. They are but slightly affected by a change in the

condition of the oil, and will consequently go satisfactorily for three years and more.

The second proportion gave M. Pierret excellent results, but he points out that (1) the smaller cylinder is rather more fragile, and (2) the insensibility to variation in the oil, which enables the escapement to go well even after it has dried up, may induce wear of the cylinder, and more especially of the pivots if the watch is not cleaned in time. And it is well known that the owner of a watch will, as a rule, not bring it to be cleaned until it quite refuses to go.

377.—LAUMAIN.—Proportions of watches that have been good timekeepers:

Watch, 36 mm. (16 lines) in diameter;
Balance, 15 mm. (0·59 ins.).—Ratio B : C :: 16·3 : 1.
Watch, 37 mm. (16½ lines) in diameter;
Balance, 14 mm. (0·55 ins.).—Ratio B : C :: 18 : 1.
Watch, 41 mm. (18 lines) in diameter;
Balance, 16 mm. (0·63 ins.).—Ratio B : C :: 16 : 1.

Referring to this latter ratio the author adds: "This balance is rather small."

We would point out that the watch is mentioned as having been well timed, and, in the absence of more complete explanations, we must suppose that this regularity was due in part to the balance-spring, and in part to the mass of the balance being so distributed that its radius of percussion was somewhat greater than is usual in a balance of similar dimensions (**354**).

378.—MM. PATEK AND PHILIPPE.—In the cylinder watches of this firm, well known from the excellent work it produces, the escapement is generally thus proportioned:

B : C :: 16 : 1.

M. Philippe, an accomplished and remarkably skilful artist, superintends the timing operations. Many of the Geneva manufacturers are disposed to employ cylinders rather larger.

379.—M. DELMAS was always able to secure satisfactory timing, in watches about 33 mm. (14 or 15 lines) in diameter, when the diameter of the wheel, with 15 teeth, closely approximated to the radius of the balance; in other words, when the ratio was:

B : C :: 16·5 : 1.

He employed the same proportion in watches 45 mm.

(20 lines) in diameter, but found some difficulty in regulating them (end of **344**).

380.—C. SAUNIER.

WATCH N° 1.—Size, 42 mm. (19 lines).

Barrel 19 mm. (0·76 ins.): measured within the teeth 17 mm. (0·67 ins.): 15-tooth wheel.

Diameter of Balance, 17 mm. (0·67 ins.)

Ratio; B : C : : 17 : 1.

The rate was maintained for several years, and was not in any way disarranged by cleaning.

The balance-spring was, if anything, rather long.

381.—WATCH N° 2.—First quality Geneva watch. 15-tooth wheel.

Balance diameter, 16·3 mm. (0·64 ins.)

Ratio; B : C : : 17·5 : 1.

The rate remained satisfactory for three years.

382.—WATCH N° 3.—Well-made Geneva watch. Size, 32 mm. (14 lines); extreme diameter of barrel 14·5 mm. (0·57 ins.) The escape-wheel projected beyond the edge of the balance by rather more than the height of the impulse plane.

Ratio; B : C : 16·5 : 1.

This watch went well for 5 or 6 years without the oil being renewed, and recovered its original rate after cleaning. Subsequently, that is, when it had gone uninterruptedly for 12 years and after undergoing a repair which was indifferently performed, it fell off in its rate. This was probably due to an increase in the friction at the depths, and inequalities in the action of a mainspring that had been carelessly handled.

383.—WATCH N° 4.—Size, 41 mm. (18 lines). Diameter of Barrel 19 mm. (0·76 ins.), measured within the teeth 17 mm. (0·67) ins.)

Diameter of Balance, 16·3 mm. (0·64 ins.)

Ratio; B : C : : 16·3 : 1.

Went well for four years, but there was a slight though constant loss.

These results all point to the conclusion that if the balance of this watch had been somewhat larger, its weight remaining the same, a longer period would have elapsed before this loss took place.

384.—WATCH N° 5.—Size, 45 mm. (20 lines).

Ratio; B : C : : 14·5 : 1.

This balance is too small; hence when the watch is carried it only remains a good timekeeper for a short interval at a time. If the spring is at all strong, it gains in consequence of the banking of the pin against the banking stud. If recently cleaned and provided with a spring that occasions an oscillation of 270°, it very soon falls off in its rate.

385.—WATCH N° 6.—Of Besançon make and good quality, made by H. Montandon. Size, over 46 mm. (20 lines).

Diameter of Balance 19 mm. (0·76 ins.)

Ratio; B : C : : 16 : 1 nearly.

The rate was good and well sustained.

386.—WATCH N° 7.—Ordinary quality.

Size, 38 mm. (17 lines). Diameter of Balance, 15 mm. (0·59 ins.)

Ratio; B. : C. : : 15·5 : 1.

The Balance was heavy as compared with other watches of this size.

For fifteen years the owner failed to get good results with this watch. Several watchmakers had examined and altered it to no purpose. One renewed the balance-spring; another the mainspring; another replaced the original balance by one of considerably greater weight, &c. None, however, were successful in securing a good rate.

It was brought to us for examination. We began by improving the depths; of the jewelled holes some were badly polished and others too thick. The action of the barrel was rendered easy and reliable, and a mainspring employed which acted uniformly and without any sudden variation. When the three elements of timing (**364**) had been properly harmonized, the watch maintained an excellent rate, notwithstanding that the balance was somewhat heavy.

The calliper of the watch made it necessary to rely on these particular corrections; had this not been the case we should have preferred to replace the balance by one that was larger and lighter, so as to diminish the sensitiveness to friction and thus secure, for a longer period, constancy in the equilibrium between these three elements by making it less dependent on uniformity of the motive force.

387.—WATCH N° 8.—Size, 34 to 35 mm. (15 lines). Diameter of Balance, 15 mm. (0·59 ins.)

Ratio; B : C : : 16·5 : 1.

The calliper of this watch was arranged by M. Leschaud, a skilful watchmaker of Geneva, and the rate remained perfectly constant during about two years; it then, without any apparent cause, became irregular.

After very careful examination we have come to the conclusion that these variations are to be attributed to changes in the motive force and in the rubbing surfaces throughout the train. These changes are explicable when we consider the small dimensions of the barrel and centre wheel, while the fourth wheel is out of proportion, and its teeth, therefore, too large. The escape-pinion, moreover, was somewhat large and its leaves too thick.

The Relation of the Diameter of the Escape-wheel to that of the Cylinder.

387a.—In paragraph **382** reference is made to the amount by which the escape-wheel overlaps the edge of the balance. Such remarks, as well as the empirical rule that the diameter of the escape-wheel should be half that of the balance, are useful as aids to examination, but must only be looked upon as giving an approximate measure.

The dimensions of the cylinder depend not only on the diameter of the escape-wheel but also on the height of the impulse plane; and this height often varies on wheels of the same diameter, and depends on the size of the watch.

TABLE

SHOWING THE RELATIVE DIAMETERS OF BALANCE AND CYLINDER IN WATCHES THAT HAVE BEEN GOOD TIMEKEEPERS.

388. Observer.	Ratio.	Size of Watch.
Gannery	B : C : : 16·5 : 1.	Ordinary dimensions.
Rodanet	B : C : : 18 : 1.	Gentleman's 18-line watch.
	B : C : : 17·3 : 1.	„ 19 „
F. Berthoud	B : C : : 20·7 : 1.	Fusee watch, 7200 vibrations.
Pierret	B : C : : 20 : 1.	Size, 14 lines; wheel, 13 teeth.
	B : C : : 18 : 1.	„ „
Laumain	B : C : : 16·3 : 1.	Size, 16 lines.
	B : C : : 18 : 1.	„ 16½ „
	B : C : : 16 : 1.	„ 18 „
Patek and Philippe	B : C : : 16 : 1.	Various sizes.
Delmas	B : C : : 16·5 : 1.	Size, 14 and 15 lines.
C. Saunier	B : C : : 17 : 1.	Size, 19 lines.
	B : C : : 17·5 : 1.	„ 19 „
	B : C : : 16·5 : 1.	„ 14 „
	B : C : : 16 : 1.	Gentleman's large watch.

Note on this Table and Concluding Remarks.

389.—The data contained in the above table, when taken in conjunction with the results at which we have previously arrived, show that in average size watches of modern construction we must not fall short of the proportion 1 to 16, and that it is generally advantageous to let the diameter of the balance slightly exceed this amount.

These figures constitute an excellent guide, but can only be regarded as final if we remember that friction, one of the elements to be considered, varies in different watches, and that a slight difference in the weight of the balance arms, or of the rim, must influence its diameter. For it should never be forgotten that the *mean diameter* is deduced from the circumference of percussion (**45**), which varies with (1) any redistribution of the mass (**354**); (2) any change in the weight of the central portion, whether it be due to the cylinder or the balance-spring collet.

390.—We will conclude with an observation (which we often have occasion to repeat) and by the mention of certain facts bearing on it.

Certain empirical rules, deduced from observation, are current among watchmakers, but they must never be accepted as *principles*. The only principles that must be followed in the construction of a machine, of whatever nature, are those fundamental laws on which the science of mechanism is based.

The timing of watches, clocks, &c., is not secured by adopting one particular form or one size rather than another, but it is mainly dependent on the arrangement as a whole, on the careful adjustment of all the parts (**270**); hence, watches that are good timekeepers are often met with having inclines high and low, straight and curved; cylinders more or less opened; long and short lifting arcs, etc., etc. The fact that a timekeeper is well regulated to-day only proves one thing, namely, that the most efficient combination of the several parts exists for the moment; but it would be rash to conclude from this that it will be maintained unless we first pay due attention to the changes which may take place in the friction and motive power.

The truth of this is daily illustrated by the numbers of watches of the very commonest quality that are found to go

perfectly well, and incompetent watchmakers cannot explain the fact. The reason, however, is very simple. Equilibrium is maintained among the several opposing influences, because one set of faults is neutralized by another set; but, owing to the indifferent workmanship, the friction changes in a very brief period, and then one set of influences preponderates, so that, in time, most of these watches, having gone well perhaps for a period of two or three years, become utterly incapable of maintaining a uniform rate.

Practical Details.

391.—The two facts mentioned below, which have occurred in our own daily experience, are noteworthy; they require no explanation.

Some years ago we sold a cylinder watch, whose rate was truly surprising. It hardly lost a minute a month. At the end of six months it was only four minutes wrong, and rather more by the end of the year. The lower cylinder pivot was broken by a fall. It was replaced with great care, but left slightly larger, and the jewel which received the original pivot, being a little rubbed at its edge, was replaced by one that had been obtained in a shop. This latter ruby was not so deeply cupped, or so highly polished as the original one. The friction of the lower cylinder pivot, therefore, was a little greater than formerly, and this increase in the friction was sufficient to occasion a slight irregularity in the going. The watch, however, continued to maintain a uniform rate, but with the loss of two, and sometimes three, minutes in the course of each month.

392.—The second observation has reference to a cylinder watch, made early in the century, which maintained its rate with considerable accuracy. All the pivots worked in brass holes, and the balance-spring was hammered, as was usually the case at that period, presenting a very unsightly appearance. This spring consisted of only seven coils; it expanded evenly and was possessed of considerable elasticity.

Irritated by its ugliness, the watchmaker to whom the watch belonged, notwithstanding our advice that he should not interfere with it, replaced this spring by a modern rolled spring, very pretty to the eye and having in the same diameter twelve coils; but he was much astonished that his watch, after going with wonderful accuracy in its original condition, went very indifferently when altered.

Its former rate was only recovered by replacing the rejected balance-spring (**350**).

TO DESIGN A CYLINDER ESCAPEMENT.

393.—In order to thoroughly comprehend the action of the several parts of an escapement, and to determine their exact proportions, one of the best means is to first draw it on a large scale showing the positions of these parts at different periods of its action.

The following is the system we generally adopt, in order to obtain working drawings of a cylinder escapement.

We will select for illustration an escapement having a 12-tooth wheel. This number is taken on account of the smallness of the engraved plates, and because the teeth are sufficiently large to avoid all fear of the lines being confused when the scale of the drawing is reduced.

We wish to know the proportions of a cylinder escapement having:

Escape-wheel, *twelve* teeth and 10 mm. (0·39 in.) in diameter;

Impulse plane inclined at an angle of 20° to its base;

Thickness of cylindrical shell equal to one-eighth the length of the impulse plane.

As the diameter of the escape-wheel is known, it is multiplied by twenty, thirty, forty, etc., according to the size of drawing required, and a diagram will thus be produced from which all the dimensions of the escapement can be ascertained by merely dividing by the above number.

The space being limited, we will only make the figure twenty-five times its natural size. Ten multiplied by twenty-five gives 250 mm. as the diameter of the wheel or a radius of 125 mm.

On a large sheet of drawing paper, fixed on a drawing board, describe with a centre C (fig. 5, plate I.) and radius 125 mm. (4·92 ins.), the arc B D Q of a circle, part of the circumference of the escape-wheel.

With a wheel of 12 teeth the interval between the heel of one tooth and that of the next will be an arc of 30°, or 360° divided by 12.

Draw the radius B *l* C. From B measure off an arc B D of

30°, or else by means of the *protractor*, supplied by mathematical instrument makers, or yet again by the more delicate methods explained at the end of the work, set off the angle B C D of 30°, and proceed thus throughout the entire circumference of the wheel.

The angle B C D exactly includes, at the circumference, a tooth and a space, and each line enclosing it touches the heel of a tooth.

The cylinder would occupy the position B A when enclosing a tooth, and when placed in a space between two teeth, it will fill the interval A D; it is thus seen that its two diameters touch at the point A, and the exact determination of this point is a matter of some difficulty.

The angle B A *g* consists of (1) the lifting angle; (2) a small supplementary angle *h* A *g*, which approximately equals 4° with a 15-tooth, and 6° with a 12-tooth wheel.

Since B *g* A is a rectangular triangle, the sum of the two angles *g* B A, B A *g* is equal to 90°. Hence, if the lifting angle together with the small supplementary angle *h* A *g* be subtracted from this amount, the remainder gives the measure of the angle *l* B *f*: draw the line B *f* which is thus determined.

In order to ascertain the ratio between the internal and external diameters, adjust the proportional compass so that the length of the short legs is to that of the long legs as 8 is to 10.

Using B as a centre with the shorter legs, and D with the longer, vary the opening of the compass until the two arcs of circles struck from these centres cut each other in a point on the line B A *f*. By this means the point A will be fixed.

The lines A B and A D will be the internal and external diameters respectively of the cylinder, and its two positions will thus be known.

Now draw the three circles *k*, *u*, *p*.

The entire drawing should first be completed with a finely-pointed pencil, for it is necessary before inking it in to co-ordinate all its parts, and more especially to slightly modify the lifting angle when this is found to be necessary.

From the centre C draw the arc x *t n* z, passing through the middle of the chords of the inclines, and the arc *h* A *r* E through the points of the teeth.

The small circles *k*, *u*, *p*, previously described, will enable us to ascertain (1) the slope of the heel of the tooth; (2) the

depth of the U-space (*v v'*). This depth should extend below the cylinder by at least half the radius of this latter.

Draw the line C A *u*, passing through the point A at which the two diameters meet, and, by dividing each arc of 30°, into which the whole circumference was divided, in a similar manner, it will be very easy to accurately and fully represent the action of the escapement in all its successive positions.

The line *m n* is drawn from the middle point *n* of an incline perpendicular to it. When drawing the curved face of the impulse curve, the point of the compass must be placed on this line.

The circumference *m i j* must now be described with a radius C *m*. It will then only be necessary to draw from the middle of each incline a tangent to this circle in order to ascertain the line on which the point of the compass should be placed to trace out each impulse curve.

REMARK.—It must be carefully observed that in this drawing no notice has been taken of the play necessary for the proper action of the escapement, and that a few degrees must therefore be added to the lift in order to prevent any loss in the lifting action from (1) the play which the tooth must have within the cylinder, and (2) the rounding of the cylinder edges.

CHAPTER V.

ACTUAL REPAIR AND CONSTRUCTION OF THE CYLINDER ESCAPEMENT.

394.—Although specially intended to assist those watch makers who are engaged in the repair of watches, manufacturers will derive from this chapter much useful information. The subject of which these latter are specially ignorant is the alteration effected by time in the condition of the mechanism of a watch. Only new watches come under their notice, and these are never seen again after they have once been exported; the manufacturer is thus seldom in a position to foresee the circumstances that will influence its going in the future.

A Word on the expression "Watch-jobbing."

395.—To thoroughly repair a watch, it must never be forgotten, is an operation which requires *more knowledge* than, and quite as much skill as, is necessary for its manufacture on

the large scale, especially at the present day when the subdivision of labour is carried to such an extreme. How then is it that the practice of repairing clocks and watches is designated by the utterly unsuitable term *watch-jobbing?*

This word, which in popular language is about equivalent to *cobbling*, has contributed not a little to bring about the discredit into which the profession of the watchmaker is gradually falling; in proof of this fact one need only mention the disdainful employment of the phrase, "He's only a jobber," by men when complaining of a watchmaker. It is to be hoped that skilled watchmakers will choose another title, another designation that is both more fair and more suitable; that they will both insist on its being used, and maintain its dignity by preventing workmen from adopting it until they have given sufficient evidence of ability.

If this suggestion could only be fully carried out, much good would result to the trade generally, and especially to those practical men who have a sound knowledge of their profession.

A Hint to Young Watchmakers.

396.—The following chapters are more especially addressed to young watchmakers. The details given are the result of a prolonged practical experience, and will help them in their daily work; this is the more necessary because, at the present day, wages are very low, and it is of the first importance that they know how to ascertain the fault, and to rapidly rectify it.

But they must always bear in mind that by the exercise of very little skill a timekeeper can be rendered *secure against stopping* by a workman of but moderate ability, the watch being at the same time ruined by the random altering to which he resorts; on the other hand, theoretical knowledge is essential in order that *timing*, the criterion of real ability, can be performed with success.

The power of systematically examining a watch, of resolving, without numberless trials, on the adjustment which is necessary in order to secure a good rate, and at the same time of feeling confident of arriving at such a result, all this is possessed by the intelligent watchmaker, but never by a man who mends on a simple rule-of-thumb system.

One need only refer to those who waste time in the practice of the less important methods and processes, and yet, even after a whole life spent in watchmaking, are often quite incapable of

making certain kinds of watch go well, or even of ascertaining why they go badly.

To Measure the Height of the Incline.

397.—The following is the method we adopt for measuring the height of the incline, a method that has been already referred to.

The number of degrees in the lifting angle is measured on the circumference of the cylinder, and, since the ratio of the circumference of a circle to its diameter is constant (**137**), it will be at once evident that the height of the incline can be represented by an invariable fraction of this diameter; it will, therefore, only be necessary to know the size of the cylinder or, in its absence, of a smooth arbor that passes with very slight play between two teeth of the wheel, in order to ascertain, from a comparison of this size with the height of the incline, the exact amount of *real* lift that the tooth will impart.

By applying this geometrical ratio of the diameter and circumference we find that:

When the height of the incline is one-seventh of the diameter of the cylinder, the total real lift is rather more than 30°;

When one-sixth, it is about 38°;

When one-fifth, it is about 46°;

When one-fourth, it is about 58°;

And for intermediate heights the lifts are proportional.

Hence, if the cylinder, or, in its absence, the arbor replacing it, be held between the longer arms of the incline-gauge (see the description in Chapter VIII., Article **503**) successively at the corresponding divisions, the distance apart of the small arms of the instrument will give the different heights of the incline, by which a total real lift of about 30°, 40°, 45°, 50°, &c., may be secured.*

This simple and convenient method has the advantage of promptly indicating the height of the impulse curve, and of at once showing how much difference may exist between escapements which to the eye are identical.

A watchmaker who accustoms himself from his apprentice-

* When making a new wheel it is possible, by merely knowing its diameter, to determine the height of incline beforehand by calculation. But since this analytical solution might, with some reason, frighten some of our readers, we shall, in its stead, give a practical method subsequently.

ship to thus measure the heights, by employing the incline-gauge, will in a short time be able to adjust them by eye, and not be very much wrong.

398.—It will of course have been gathered that the height of the incline is always the space contained between the heel of the tooth and the circumference passing through the point.

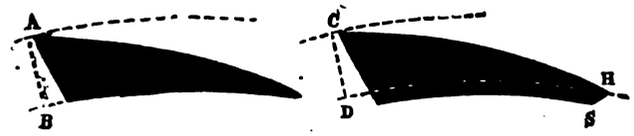

Fig. 28.

Thus, with a tooth formed as A (Fig. 28), this height is equal to A B, whereas it is only C D for a tooth shaped as H S D C.

It would be a useful precaution if, where this latter form is adopted, a line such as H D were allowed to remain on the flat of the tooth; this line would merely be a portion of the circumference that passes through the points of the teeth, that is, through the points of contact with the resting surface.

Practical Details concerning the Escape-Wheel.

399.—We have already shown the inconveniences which resulted from the brass wheels formerly employed in this escapement. In more modern times they have been replaced by wheels of steel, and these can be both light and thin. The hardening, since it increases their rigidity, avoids all risk of the teeth being bent, and the decomposition of the oil is rendered less rapid; further, if the steel is of the very best quality, carefully hardened, and if the points and inclines of the teeth are highly polished, experience has shown that wear does not occur, and that the pressures on the edges of the cylinder are maintained constant throughout a very long period.

In workshops the size of the escape-wheel is determined by that of the fourth wheel of the train (that carries the seconds hand), a slight allowance being made to ensure the heel of the tooth clearing the pinion of this wheel. Many Swiss makers unwisely make the fourth (or seconds) wheel as large as the third or very nearly so, in order, as they affirm, to increase the size of the escape-wheel, and thus, while rendering the friction during rest less detrimental, to increase the distance between the balance and the pinion of the fourth wheel and be enabled to introduce a much larger balance: a very erroneous argument,

which is apt to mislead from apparently having a theoretical basis. The sizes of the wheels must, as we have already said, diminish gradually in proportion to the force by which they are impelled; and it must never be forgotten that, of two extremes, *it is better to have the escape-wheel somewhat small and a heavy regulator rather than a large wheel in conjunction with a light balance.*

400.—For small watches (under 30 mm. or about 13 lines) the escape-wheel should be less in proportion, for, in watches of this class, (1) the number of its teeth is less; (2) the motive force is very feeble, and if the last wheels are large they oppose too much resistance to motion; as a consequence a very slight change in the consistency of the oil suffices to disarrange the timing; (3) the number of oscillations per hour is greater (19,000 to 21,000) so that the movement of the cylinder is more rapid.

The teeth should be sufficiently numerous. A large wheel with few, and therefore long, teeth requires a large cylinder, which will be more influenced by variation in the motive force (**345**).

Only the portion of the wheel from the centre to the pillars must be tempered to a greyish-blue tint. If the teeth could remain as they are left from the hardening, it would be all the better, but this is impossible; the more they are let down the less satisfactory they become. It is hardly necessary to add that the wheel must run true and flat; that all its teeth must be made true throughout their length, and as regards the distance between the point of each tooth and the heel of the one which succeeds it; that its arms should be narrow so as to reduce the weight; that the pillars must be rounded to a sufficient depth below the level of the incline lest they rub against the cylinder at the same time as it does; and, indeed, in order to prevent them from drawing away the oil which by its more rapid thickening would thus tend to paralyse the motive force.

The acute angle of the heel should be very slightly rounded and polished; the points should be *rounded off in both directions* so as only to rub in the middle.

The inclines and points must be very highly polished, but *fine grinding* is preferable for the flat of the tooth, since it facilitates the retention of oil, an important point in this escapement.

The U-arms should be rather narrow and placed near the heels of the teeth, as shown in figures 2 and 5 (plate I.), otherwise in the large arcs of vibration the end of the banking slot would touch against these arms and interfere with the going of the watch.

It is well to cut the heel in a sloping direction, as is shown in the same figures, especially when the height of the tooth is considerable.

Construction of the Escape-Wheel.

401.—The explanations which this question involves are very numerous and rather go beyond the repair of watches. We have therefore devoted the whole of Chapter VII. to details of construction of the cylinder escapement. The reader will find there and in Chapter VI. *(Causes of stoppage and variation)* all the information he can require either for constructing an escape-wheel or adjusting one already made and such as is met with in a watch or at the material dealers.

In addition to all the details of construction, there will there be found all necessary particulars with reference to fixing the wheel in position.

To Make the Escape-wheel True.

402.—This operation consists in the equalizing of the internal and external drops.

The method usually adopted when the inside drops are unequal is to adjust all the teeth to equal the smallest, which should just enter into a hole in a small brass gauge made on purpose. This hole then serves as a measure for the other teeth.

To adjust the U-spaces they should, in the absence of a better means, be measured with a smooth cylindrical arbor passing without play between the point and heel at the greatest space.

The matter in excess is removed by an iron file charged with oilstone dust and oil; a better means, however, is to employ a small ruby or sapphire file.

This method will not always procure an absolute equality in the drops; for to guarantee this it would be necessary that the gauge embraced a space and a tooth, but the above will suffice for all ordinary purposes.

It is preferable to reduce the teeth at their points rather than at their heels (**411**).

Various tools for making horizontal escape-wheels true can be obtained at the tool shops. Some of these will be found described in the article devoted to this subject in the *Guide Manuel.*

Practical Details on the Cylinder.

Ruby Cylinder.

403.—A cylinder that is made with ruby, providing it is supplied with oil, is very nearly indestructible, except in case of accident. We have nevertheless occasionally met with them in which these working surfaces were deeply worn; this is doubtless caused mainly by the injurious employment of Balas or Spinel rubies in place of Oriental rubies or sapphires, which are the only ones possessing the requisite degree of hardness.

The friction against a ruby cylinder is less harsh than that due to the action of steel against steel, but it has the disadvantage of rendering the timing difficult, mainly from this very fact of the friction during the rests being less.

Those who advocate the abolition of friction will, of course, at once cry out that the preceding paragraph is paradoxical; but will they explain the following fact, which is well-known to every watch-jobber? As a rule it is only possible to regulate escapements with ruby cylinders after the lapse of a few weeks or even months, and this precisely corresponds with the period at which the consistency of the oil becomes greater.

In our opinion the fact admits of a very simple explanation, and all that we have hitherto said may be summed up in a few words.

404.—The proportions adopted in the great majority of escapements with ruby cylinders have been simple copies from those with them of steel. It follows therefore that the ratio existing in the first instance between the impelling force and the resistance caused by pressure on the resting surfaces (that is the friction) is destroyed. The principal elements could then only be brought into accord by adjusting afresh the several parts of the escapement in such a manner as the diminished friction shows to be necessary (**364**).

The disuse into which the ruby cylinder has fallen is due, not only to its high price, but, perhaps, in a still greater degree, to errors in its construction. Indeed we have met with many escapements of this class, which proved to possess one or more faults, such as the following:

(1) The dimension of the axes excessive; (2) too great a a weight at the centre of the balance, as compared with the weight of its circumference; (3) the balance not running true; (4) two banking pins being necessary, or perhaps one very large one; and this limits the arc available for the vibration of the balance, often occasioning banking when the oil is fresh.

A workman, when about to replace a ruby by a steel cylinder, must first be careful to ascertain, by the method which we have explained in considerable detail, that the balance is well adapted to the escapement. In the great majority of cases of this kind that we have been called upon to examine we have found the balance either too small or too heavy when a steel cylinder is substituted for one of ruby.

Chapter VIII. will enter into some particulars relating to the construction of a ruby cylinder.

Steel Cylinder.

405.—A cylinder merely made of steel, thoroughly hardened where the friction occurs, will, providing it has been carefully made and well polished, continue for a very long time in action without showing any signs of wear; but, as in the case of the ruby cylinder, it is very important that oil be supplied in sufficient quantity.

The steel cast in square bars and known as forged steel is considered by most authorities to be the soundest; and yet nearly all the French and Swiss watchmakers employ the *drawn* or *screw* steel, because they consider it to be less liable to crack than the forged steel. We prefer this latter, because, being more homogeneous, it takes a better and more even polish than any other, but it has the disadvantage of being more easily burnt if a cherry red heat is exceeded in the hardening.

The admirable cylinders, still in excellent condition, that are met with in the older horizontal watches, were all made of forged steel, but this metal requires so much care, both in working and hardening, that it was wisely replaced in the factories by drawn steel. We shall explain the reason of this in the article which treats of the construction of the cylinder.

Formerly the plugs were formed of copper, a fine steel wire being centred in them so as to form the pivots. At the present day the plug and pivot are turned out of the same piece of steel, which is first hardened, and then let down to a deep blue shade.

The finished cylinder should pass with but slight play between the point and heel of two successive teeth. Its internal diameter will depend on the length of a tooth, and the play should be rather more than that which is allowed outside.

Should the tooth be too short, the shell of the cylinder would of course be then very thick. This would materially increase the width of the lips, and the inequalities in the friction during the internal and external rests, because these two rests would occur at the extremities of radii of different length. When the thickness of the shell is excessive, a watch usually goes very badly, and it becomes necessary to replace it.

The thickness must not be more than that which is necessary in order to ensure due solidity. One-eighth of the length of the inclined plane must be taken as a maximum limit, and at times it may be less without inconvenience.

A too open or too close cylinder.

406.—It has been demonstrated that the amount of the cylinder opening is not a matter of indifference; that the exact depth to which it is to be cut away is indicated by the manner in which the effect of the friction during rest is neutralized. We moreover observed that, commencing with a cylinder opened to the requisite extent, and gradually increasing this opening, the extent of the arc of oscillation of the balance as gradually diminishes; and, in the converse case, if the original cylinder is replaced by others more and more closed, the thickening of the oil renders the escapement the more sluggish in proportion as the length of the period of rest becomes more considerable.

This fact having been made clear, we would observe that, as regards the cylinder escapements met with at the present day, where the half-shell measures from 196° to 200°:

When the cylinder is too much *open* the extent of the arc of vibration is not sufficient. By making the escapement deeper we increase the external drop at the expense of the internal, and the escapement is rendered still more unsatisfactory. As to a very *closed* cylinder, it involves an impulse curve more inclined than would otherwise be necessary, and therefore a greater motive force must be applied in order to ensure the watch against setting; but, if the height cannot be thus increased, the point of the tooth will act against the back of the edges, or even against the resting surfaces, and butt against them. It will be essential, in order to facilitate the

going of such an escapement, to make it somewhat shallower, and, since the middle of the incline will then not pass through the centre, the external drop will be increased, while that within the cylinder will have become shorter.

Measure of the Opening.

407.—It has been already seen (**340**) that in cutting the cylinder opening rather over *five-twelfths* of the diameter is cut away, that is to say, when the cylinder is finished the depth of the half-shell should be somewhat under *seven-twelfths* of the diameter, or one-half plus a bare twelfth.

The height of the opening should be about *one and a half* times that of the wheel, measuring from the base of the pillar to the upper face of the tooth (see the end of 6: *To make a cylinder*, article **464**, Chap. VII).

The edges and resting surfaces of the cylinder must be very highly polished, and great care is necessary to avoid sharp cutting angles or burrs, especially at the inside end of the slope of the disengaging edge; such a burr constitutes a cause of stoppage that is but seldom noticed.

The tools employed for measuring the amount of opening will be described in Chapter VIII.

Depth of the Banking Slot.

408.—The small *banking slot*, whose object is to enable the balance to perform a complete rotation, must measure 2[illegible] of circumference, so that the pillar will not measure more[illegible]n 90°. Sometimes it is even cut deeper, when the U-arms a[illegible]ot level with the heels of the teeth, or when the teeth of the w[illegible]el are too short, so as to avoid the striking of the end of this slot against the U-arms.

In such a case, the portion removed exceeds three quarters of a circumference by the amount by which either the teeth are too short, or the heels project beyond the U-arms. An iron file charged with oilstone dust and oil is generally employed for this operation, or else a small flat ruby file held edgeways.

The height of the banking slot is about three times the thickness of the flat of the escape-wheel. We say *about* this amount, because it is not sufficient with very thin wheels, and too much in the contrary case.

When the slot is cut in the manner above explained, the balance should perform an entire revolution without being

touched by the U-arm; if it does not do so the banking pin must be out of place. We shall subsequently see (**425**) how such a fault can be counteracted.

A too large or too small Cylinder.

409.—We shall not here consider the size of the cylinder from the point of view of its regulating power (**261** and **269**), but simply the relation of its internal and external diameters to the escape-wheel in the escapements ordinarily met with in practice.

If the inside and outside drop are insufficient, either the shell is *too thick* or the teeth of the wheel are *too long*.

If there be too much drop inside and not sufficient outside, the cylinder is *too large*. With too much drop outside and an insufficiency inside the cylinder is *too small*.

In either of these two cases, or when it is too thick, the cylinder must be replaced.

We would, however, observe, for the benefit of watch-jobbers, that when a cylinder with a shell of proper thickness is somewhat large, or when a cylinder with a too thick shell is re-made, it is nearly always possible, by adjusting the outside drop, without specially troubling about the inside drop, to ensure sufficiently accurate timing (providing the watch is not otherwise faulty).

With too much internal drop and none at all outside, the shell being somewhat thick, the teeth are too short and the cylinder too large, and with two such faults combined it becomes necessary to re-make the escapement.

General Rule for Watch-Jobbers.—When the external drop is slight, even although it is considerable within, the shell thin (one-eighth the length of the impulse plane as a maximum), the cylinder sufficiently closed, and the tooth of the right inclination and not too much curved, the vibrations will, as a rule, be free, and pretty good timing can be secured.

It must be clearly understood that this observation only applies to the *repair* of watches, for every new escapement must, if it is expected to maintain a good rate, be constructed in accordance with the principles above laid down.

Verification and Adjustment of the Drops.

410.—The objections to an excessive drop are already known; it occasions a loss of force and increases the difficulty

of timing; we must, then, allow of no more than is essential to ensure the play of the escapement.

In order that the drop may not exceed this necessary amount, it is of the first importance, as we have already said, that the middle of the chord of the incline pass through the centre of the cylinder and that a very thin cutter be employed in forming the teeth of the wheel. In practice it is usual to give rather more drop inside than outside; for it is almost impossible to be certain that the middle of the incline is not somewhat short of the centre or beyond it, in which case the heel and point of the tooth are liable to rub at the same time, in spite of a slight latitude in the internal drop: the same effect might be observed on making the escapement somewhat deeper.

Such a fault might also be occasioned by the operation of polishing having affected the trueness of the internal cylindrical surface.

The drops will be equalized by trueing the wheel (**402**). At the same time, if the irregularities in the drop are considerable, and one is unable to replace the wheel, it will be well to give just sufficient drop for the longest teeth, rather than to have excessive drops throughout, and the following is the method of procedure:—

The mainspring having been wound up, and the balance held by a strip of paper, it is caused to rotate by means of a fine piece of peg-wood, until a tooth falls to rest. Then, causing the balance to travel a little backwards, it will be easy, by giving the wheel a little oscillatory motion, to ascertain whether it has the necessary play both when a tooth rests within the cylinder and when this latter is held within a U-space. After repeating this operation round the entire wheel, and making a mark with rouge opposite each U that is too narrow and each tooth that is too long, the wheel is removed and the requisite corrections made with a small ruby file. The teeth are then re-formed and polished carefully where necessary.

411.—It is preferable to reduce the tooth at its point, unless, of course, the heel projects too much beyond the arm of the U, or the inclination of the impulse plane is too great. As a rule, either of these circumstances will render it necessary to carefully touch the teeth of the wheel. We have already said sufficient to enable the watchmaker to decide as to the best method to adopt in such a case.

412.—*Observation.*—The drop, when the incline is curved, is the more violent as the curvature is more pronounced. In some cases, which the workman will be able to decide upon for himself, it is well to work down the impulse curve, an operation which can be performed without difficulty by means of the *Incline-Tool* described in Chapter VII. The incline should still project considerably in front of the pillar after it has been thus re-formed.

Escapements that we have corrected in the manner above explained, were at once characterized by (1) an increased amplitude of oscillation; (2) a less amount of drop, for the noise heard when it occurred was very sensibly reduced.

The Lift of the Escapement.

413.—Starting with a very slight lift, we know that, if the pitch of the wheel within the cylinder be gradually increased, there is a sensible increase in the extent of the vibrations until the middle of the chord of the incline coincides with the centre of the cylinder; but, if this point be exceeded, the extent of vibration either remains the same or diminishes (**311-313**).

The entire question of the lift then reduces itself to this one rule: *the middle of the chord of the incline must coincide with the centre of the cylinder.* The escapement is then *in adjustment.*

Relying on this principle, we would observe that (1) by *deepening* the lift, that is by pitching the incline beyond the centre, the period of rest is increased, the liability to setting becomes greater, the external drops too great, and the internal too small; (2) by falling short of the centre, the oscillation is of insufficient extent, the external drop too great, the internal too small, and, if the opening of the cylinder is considerable, the points of the teeth fall on the lips, instead of the resting surfaces, etc.

The mistake made by those who expect to increase the energy of impulse by deepening the lift is now manifest; the only effect of such a practice is to introduce an impediment and source of irregularity from the very first. An escapement so arranged is sure, in the course of a few months, to move sluggishly; it will be very liable to set, and the rate will soon become variable.

If, with an escapement in which the middle of the incline is placed centrally, the vibrations are not of sufficient extent, it

results either from the escapement being constrained or badly made, or else from an insufficient motive power; it is generally due to bad depths, a bad mainspring, too low impulse curves, too great cylinder opening, etc. (**343**). As already explained, we can practically ascertain whether the incline is in adjustment by this criterion: it is the point at which the external drop is a minimum.

414.—When the half-shell measures 196°, if four impulse planes be made, whose heights are 1/7th, 1/6th full, 1/5th, and 1/4th the diameter of the cylinder, the centres of the inclines will coincide with the centre of the cylinder when their *total apparent* lifts are respectively about 30°, 35°, 40°, and 45°.

If the cylinder be a few degrees more open, the number of such degrees must be deducted from the amount of the lift; if, on the other hand, it be more closed, the amount by which the half-shell exceeds 196° must be added to the lift, in order to ensure that the middle of the chord of the incline may pass through the axis of the cylinder.

It is a good practice to draw, on a large scale, a number of half-shells of 196°, in conjunction with impulse curves of different inclinations, and to indicate against each the amount by which the real exceeds the apparent lift. These designs will then, so to speak, serve as tables of reference.

Chapter VII. is specially devoted to details of construction of the cylinder escapement, and in it will be found all necessary information bearing on the making of the cylinder, as well as the pivoting and adjustment in position.

Pivots and Axes.

415.—In every escapement it is essential that the balance be perfectly free, in order that the arcs described may be of sufficient extent; thus the diameter of the pivots must be as small as possible, due regard being had to solidity.

As a rule their diameter is about one-eighth or one-tenth of that of the cylinder. The only effect of exceeding this amount is an increase in the friction, and a consequent diminution in the extent of the vibrations.

In practice the length of a balance-pivot is about three times its diameter. The extremity is made flat with a sharp angle, only rounding it so that it just does not scratch on the

nail. This form is adopted so as to render more uniform the total friction in different positions. It will be subsequently seen, however, that such a practice is not always beneficial.

The shoulder must have an elongated conical form. This shoulder after all is nothing but a useless mass of metal which attracts the oil and, when this has become thick, increases the resistance opposed to the balance, since there is a considerable surface constantly exposed to its action.

For the same reason it is essential to have hollows against the cylinder, and the axes should be rather long, so as to prevent the faces of the plugs and shells from attracting the oil. When space is available the lower shell and axis should be proportionately longer : the plug will then be more solid, and, the pressure acting on the cylinder being more equally divided between the two pivots, the friction will be less.

All the escapement pivots should work in rubies of good quality, whose holes, not being liable to increase, will accurately maintain the positions of the several parts. If they are well polished and kept supplied with oil, the friction is less than in brass holes and the pivots resist wear longer ; but it is important that these pivots be made of hardened steel, tempered only to a deep blue and that they be round, cylindrical and carefully polished.

The play of the pivots in their holes should be, according to the experiments of reliable authorities, a sixth of the diameter of the pivot. Thus, if a pivot placed between the jaws of a micrometer marks 6, the hole should be of such a size as to admit, without play, a pivot marking 7. Every apprentice should possess a carefully made pivot gauge, and a number of blue steel wires at the extremity of which to make pivots of the sizes of the several holes until he has attained the requisite amount of skill to enable him to correctly adjust the play by means of the eye and touch; that is, by observing what amount of shake of the pivot in its hole is possible (**417**).

The pivots of the escape-wheel are made of the same size as those of the cylinder, sometimes they are rather finer, and it is sufficient to make their length twice their diameter.

416.—A very serious fault, observable in the majority of horizontal watches, is the almost entire absence of any axis to the pinion of the escape-wheel. Such an omission causes the oil from the pivots to spread itself over the pinion, which

attracts it; this increases the wear of the parts in contact, and makes the friction of the depth depend upon the greater or less fluidity of the oil as well as on the amount of dust incorporated with it; hence this pinion is usually found to be worn or pitted after going a very short period, and many watches vary, and even stop, simply from the thickening of the oil on its leaves.*

If the great majority of watchmakers, or those who call themselves such, instead of perpetually indulging in empty platitudes, would make purchasers understand that the conditions which are essential to ensure good timekeeping cannot be secured in watches of less than a certain thickness, one might then, in addition to securing other advantages and without making the thickness in any way excessive, retain the two axes which are essential to the escape-wheel pinion. Their importance will be the more manifest when it is remembered that the escapement pinion is the last portion of the train, and is thus acted on with the least energy by the motive power.

Since the period at which we wrote the above in our first edition matters have somewhat changed.

At the present day the public is utterly disgusted with thin watches; but there are circumstances which make us fear that, under the influence of inferior watchmakers and ignorant manufacturers, who manage to overcome their difficulties more easily with thick watches, the public will be liable to go to the other extreme.

We are constantly meeting with watches of considerable thickness, produced in certain factories, and yet so badly made that the escapements are characterized by nearly all the faults met with in thin watches.

Each axis of the cylinder should be formed like a double cone, one of the cones being inverted (Fig. 1, Plate I.); by such an arrangement the oil is prevented from reaching the chamfers of the plugs (**91**).

The axis of the escape-wheel pinion should have the same doubly conical form, so that the oil, which must be supplied very sparingly, may be retained on the pivot. If ever so

* The presence of oil on the leaves would not occasion wear if it continued pure, but it collects all the dust and dirt on the points of contact; a sort of fine sand is thus spread over the surface, and under the pressure of a depth with such engaging friction as that of a six-leaved pinion it acts in a manner precisely analogous to oilstone-dust and oil.

slightly in excess, it spreads and reaches the pinion, which is then covered with oil, while the pivot-holes are dried up.

These axes of the escape-wheel must not be too large. The cone contiguous to the pivot has a double effect; it retains the oil and reduces the extent of the shoulder. It must always be narrow, and especially so in small watches.

Jewelled Holes.

417.—The rubies or other jewels that receive the escapement pivots must be perfectly straight; in other words, they must be so fixed that the axis of the hole is perpendicular to the plate. Otherwise the pivots, being held on an inclination in the jewel-hole, are liable to become worn all the sooner, and to be cut even in large holes. It is important to make sure that in the side and edges of the holes there are no scratches or cracks in which the fine diamond dust might have gathered. It would mix with the oil and very soon destroy the pivot.

It is necessary that there be a very slight interval between jewels and the faces of endstones, in order that, through the action of capillarity, the oil may be gradually supplied to the pivot-hole as the small quantity contained in it gets dried up (**91**).

If the holes are too large, the total friction becomes variable in consequence of the several parts being differently affected by a change of position (in which case the timing in positions becomes a matter of extreme difficulty), and the oil soon spreads over the endstones. All these influences taken together have a very prejudicial effect on the rate.

When the holes do not allow of sufficient play to the pivots, the balance may be free enough when the oil is fresh, but as soon as this becomes thick, the vibrations are impeded and the watch either varies or stops; for, be it remembered, it is necessary to give more play, in proportion, to the cylinder pivots than to those of the other mobiles, in order to ensure that the balance is always free.

The holes should have oil-cups of suitable depth; by this means the friction is reduced.

In a vast number of modern watches, and unfortunately even in many of those which pretend to be of a superior quality, the jewels are rather a *blind* than in any sense beneficial. Badly worked, and of insufficient hardness, they are less serviceable

than good carefully hammered brass presenting a sufficient extent of rubbing surface to the pivot (**41**).

Our treatise on *Jewels* (published separately) enters into full details on this subject, and we will conclude with one important observation.

418.—Some authors have stated, *a priori*, that by increasing the number of jewelled holes, the timing is materially facilitated.

Now observation shows that, in cylinder watches constructed thirty or more years ago, the timing becomes a more and more delicate operation as the number of jewelled holes is increased. Generally speaking, with eight rubies such watches are characterized by a greater sensitiveness to changes in the condition of the oil and in the temperature, than when all the pivots of the train work in brass holes.

The explanation of these phenomena is not far to seek; it is entirely a question of friction.

By substituting hard rubies, with their holes well cupped and highly polished, for the brass in which the third and fourth wheels work, the transmission of the motive force is facilitated. It acts then more promptly and energetically.

It must, therefore, necessarily follow that the irregularities in the motive force due to the depths of the initial mobiles, and the occasionally irregular movement of the mainspring as it uncoils, have a much more pronounced effect on the escapement, and the harmony existing amongst the three principal elements of the arrangement is at every instant being disturbed, &c. (**364**).

The following observation has a bearing on this subject.

419.—Those modern watches, jewelled in eight holes, which have a good rate, are nearly all provided with cylinders rather smaller, and balances heavier, than are found in older watches of the same dimensions.

We would mention one fact having reference to a gentleman's watch 40 millimetres (18 lines) in diameter. The movement had been made by a very clever maker of small watches; either from habit, or with a view to showing good workmanship, he had made all the pivots of the train unusually fine.

For several years this timekeeper, which was unquestionably well made, went very badly. A watchmaker, doubtless suspecting the cause of these irregularities, removed the four

rubies from the third and fourth wheels, and replaced their pinions, making the four pivots of the usual sizes. After these alterations the watch became a reliable timekeeper.

Practical details on the Balance-Spring.

420.—In the modern callipers the diameter of the hair-spring is equal to the radius of the balance. With the compass set to half this radius, and with the cock pivot-hole as a centre, make a mark on the stud, and another under the projecting end of the index. The second of these, passing between the two regulator pins, indicates their position. The mark on the stud gives the point at which the hole should be drilled to hold the extremity of the spring.

Within this space are generally included from eight to twelve coils.

It is of course evident that this empirical condition cannot be in any way regarded as precise. It is, however, the rule most generally adopted in practice.

Much has been written on *long* and *short* balance-springs, without throwing special light on the subject; attention has, however, been drawn to the fact that, when they are too long, the timing of cylinder escapements is less satisfactory. A spring of about eight or nine turns seems to be the one best adapted to the generality of such escapements, and many well-known watchmakers, after a prolonged practical experience of this escapement, have come to the conclusion that, in the great majority of cases, the timing is worse as the length of the balance-spring exceeds the amount above given.

The length of the spring adapted to any given cylinder escapement should bear a certain relation to the velocity and the extent of the supplementary arc, as a clever observer, M. A. Vallet, has noticed. We would add that there is yet another condition to be satisfied, for the isochronism of the balance-spring of a chronometer is not identical with that of an escapement in which the rest is frictional.

This question will be considered in its proper place; it cannot be discussed in the present chapter, which is intended as a summary of the general principles that guide experienced practical men. We will only observe that, in the great majority of well-made cylinder escapements, *there is one length of balance-spring* that secures the best and most permanent timing (**350**).

421.—A balance-spring, when pinned in position, should be so situated that the balance has to traverse equal arcs on either side of its position of rest, in order to allow of the entrance and exit of a tooth.

When the balance-spring has to be selected, the external coil is held in the tweezers, and the internal extremity is hooked into the slot of the cylinder, which can thus be lifted together with its balance. This process is termed *weighing* the balance-spring, but the expression is objectionable. Its *weight*, or rather *strength*, is approximately in proportion to the number of vibrations required per hour when, on lifting the balance in this manner, the length of the cone formed by the spring is about one and a half times its diameter, the several coils being equidistant and thoroughly elastic. But on such a subject a book cannot replace the practical instruction of a good master. In addition to this indifferent method, several other more certain processes will be found in subsequent articles headed *To Time a Horizontal Watch Expeditiously* (**432-438**). The instrument employed for measuring the strength of a balance-spring will also be described in its proper place.

A hairspring must be circular and accurately centred; that is to say, when attached to the stud on the cock, and with the external coil between the regulator pins on the index, the centre of the collet pinned to the inner coil must be in the axis of the cock pivot-hole.

Its elastic force must be the same throughout, in order that its entire length may vibrate and that the coils may open and shut together without touching. When such is not the case, we conclude that the coils are not of equal strength, and it becomes necessary to replace the spring; for, as we have already explained in considering the verge escapement, *the uniform and perfect expansion of a balance-spring is the best test of its regulating power* (**180**).

It must rest perfectly flat throughout its action, remaining parallel to the cock, and there should be sufficient distance between this and the balance arms to ensure freedom of action to the spring. The external coil must not strike against the stud or the centre wheel. It should lie in the path traced out by a point midway between the regulator pins of the index; for, otherwise, it might happen that in certain positions these pins strain it, and by moving the index towards slow we might find the watch to gain.

In all positions it must be quite free between these pins, which should be carefully dried, in order to prevent stickiness of the outer coil. They must not be made of steel, as it is very liable to rust and to become magnetic.

The second turn must not strike against the innermost pin.

The index is provided with a small brass block called a *turn button*, which can be caused to rotate and is provided with a thin projecting strip to prevent the balance-spring from slipping from between the pins.

Lastly, very great care is necessary in the selection and fixing of the spring, for it has a most important influence in the timing.

In jobbing, if a spring be found to be too weak, its strength should be increased by removing a little from the centre, as the coils there are more sensitive than those outside. When the strength is excessive it can be reduced, but this must be done to the same extent throughout its entire length, an operation of some difficulty, as will be gathered from the subsequent description of the methods usually employed. It is, then, preferable to replace it by a spring of suitable strength.

Prior to putting an escapement together, the workman should make sure that the collet holds firmly on the balance, and that it is accurately in the centre of the spring. (See the article on the *Balance-Spring* in the Third Part of this work.)

Practical Details Relating to the Balance.

422.—In factories it is customary to take as a measure of the diameter of the balance, the diameter of the barrel cover, when this cover is on the same side as the teeth ; this amounts, very approximately, to the external diameter of the barrel drum itself.

This measure may be adopted, since in modern watches of average dimensions it only differs slightly from the strictly correct diameter ; but this adoption of it must be dependent on our experience of watches that are well timed for all positions and comparable in every respect with the one we have in hand.

Certain makers, who from long experience have observed that good timing always follows on the securing of a certain proportion between the radius of the escape-wheel (on which that of the cylinder depends) and the radius of the balance, have fixed as a *maximum size* of this latter *twice* the diameter of

the escape-wheel, and, as a *mean, twice that diameter less a tenth.* This so-called rule cannot fail to give variable results. We have already had occasion to refer to this fact in paragraph **387a.**

We shall revert to the subject in the article devoted to the consideration of the *Annular Balance*, but would at once observe that, when endeavouring to ascertain the correct size of the balance, the diameter that renders the escapement sufficiently insensible to variations in the motive force constitutes a satisfactory approximation (**264**).

423.—As to the weight of the balance it must depend upon the actual impelling force of the escape-wheel. Now, although we may not yet possess accurate information on the subject, we are able to employ two methods for indicating the energy of this impulse:

(1) By observing the greater or less facility with which the escapement commences its motion when the mainspring is wound up to a definite extent (**329**);

(2) The time required for the balance to acquire its mean amplitude of oscillation, reckoning from the commencement of its motion due to the winding up of the watch. In some watches it may even be as late as the *fifteenth* vibration that the balance acquires its average maximum extent of oscillation. (See paragraph **440** and the general chapter on Balances.)

Since a moderator is more efficient when slightly heavy, providing the weight is in no part excessive, it is advisable when experimenting with the balance that it be somewhat heavy in the rim in the first instance; its subsequent diminution should depend on the results of the above observations numbered (1) and (2), and we must remember that:

Too light a balance will never regulate efficiently; it is too much at the mercy of the motive force, the resistance of the air, the clogging of oil, &c. A balance that is too heavy increases the harshness of friction and the risk of breaking pivots with a fall. It requires a stronger balance-spring, a greater motive force, and at times renders the watch more difficult to time, especially in positions.*

* A heavy balance increases the difficulty of timing in positions. When it is placed horizontally, the balance pivots are subjected to less friction; for this only occurs at the end supporting the weight; if timed for the vertical position it would gain when horizontal; conversely, when vertical the friction of the balance pivots increases, since they rub throughout their entire length against the sides of the

In paragraph 354 sufficient details are given as to the best disposition of the total weight of the balance.

The most suitable material for the construction of the balance is one which, while sufficiently firm, contains the greatest weight in the least volume. Steel is unsatisfactory, notwithstanding that, after hardening, its firmness is greater than that of any other metal; for, besides being liable to rust, it is apt to become magnetized; and, volume for volume, it is lighter than brass.

A balance should be made of well-hammered metal; it should be perfectly free and truly poised, and its form should be such as will cut the air with the greatest facility. It should be perfectly true and in one plane, nearly all its weight must be at the circumference, for it is essential that the centre and arms contain a minimum of matter. It must be free of the cock, the centre wheel, the balance-spring stud, the curb pins, the balance-spring itself, escape-wheel bar, plate, etc. Care should be taken when riveting it in position that one of its arms is exactly over the back of the cylinder, so that, if necessary, the escapement may be examined on the depthing-tool.

We shall explain the special precautions to be taken in making a balance in the *Watchmakers' Handbook.*

Number of Vibrations per Hour.

424.—The amplitude of the oscillation of a balance in a horizontal watch is usually about three-quarters of a complete circle (270°), for, as long arcs secure the more perfect timing, the extent should be increased as far as the risk of banking will permit.

Make the escapement so that it gives 18,000 vibrations per hour (5 per second). As this number of vibrations gives good timing it is generally adopted (but not in those subject to severe shaking) except in the case of those that are under

pivot-holes, and the watch, if timed for the horizontal position, will lose when hanging vertically.

This variation in the friction of the balance pivots with a change of position always increases with the weight of the balance, as may be easily proved. (*Jurgensen.*)

These facts are all perfectly true in the generality of cases, but, when the pivot-holes are carefully made and properly cupped, the friction of the pivots can be rendered independent of position. It is also possible to modify the gain or loss by using another balance-spring that is better proportioned.

29 millimetres (13 lines) in diameter. These small mechanisms, owing to the very slight motive force, are specially affected by being carried and by changes of temperature, and should therefore beat from 19,000 to 21,000 and even 24,000 per hour, according to their minuteness. It does not render them at all more liable to wear.

We give the number of oscillations adopted in Geneva in accordance with the size of the watch.

Diameter of Plate.	Number of Vibrations.
From 45 to 29 mm. (20 to 13 lines) . .	From 17,000 to 18,000
27 mm. (12 lines)	18,000 „ 19,000
22 to 23 mm. (10 lines)	19,000 „ 20,000
20 to 13 mm. (9 to 6 lines) . . .	20,000 „ 24,000

The large size watches, 45 mm. (20 lines) in diameter, are timed very well with 18,000 vibrations per hour; but this is only on one condition, which, without being invariable, is at the same time nearly always realized in practice: namely, that the diameter of the balance is less than that of the lid of the barrel, when, as is usual in modern watches, the barrel occupies the entire space between the centre pinion and the edge of the plate.

Any watchmaker who is cognizant with the principles of mechanics will easily understand the reason of this.

Over-Banking of the Balance.

To fix the Banking Pin and Stud.

425.—The balance in a cylinder watch is capable of performing a complete rotation.

If the vibration exceed this amount, it will be found that, on turning from right to left,* the end of the banking slot of the cylinder, represented by the dotted line *o d* (A, fig. 29), will strike against the U-arm, the tooth, projecting beyond the engaging lip, will become wedged and the cylinder will be held fast so that it cannot be brought back by the balance-spring; such a condition of things is indicated in figure A.

This is what is called *over-banking*.

Over-banking would also follow on the cylinder being turned too much from left to right; for if the tooth were no longer held by the locking surface it would fall inside the cylinder; the U-arm would drop against the end of the slot, and on the cylinder being brought back by the balance-spring

* Or in the opposite direction if the escapement be left-handed.

it would butt against the heel of the tooth, the end of the slot at the same time being held against the U-arm, as is shown at B and C, figure 29.

Over-banking is avoided by fixing in suitable positions the cock banking stud and the banking pin of the balance.

When the escapement is at rest and uninfluenced by the mainspring, that is to say, when the mark on the balance is directly over the central mark on the plate, the banking pin should be opposite its stud and in precisely the same plane with it.

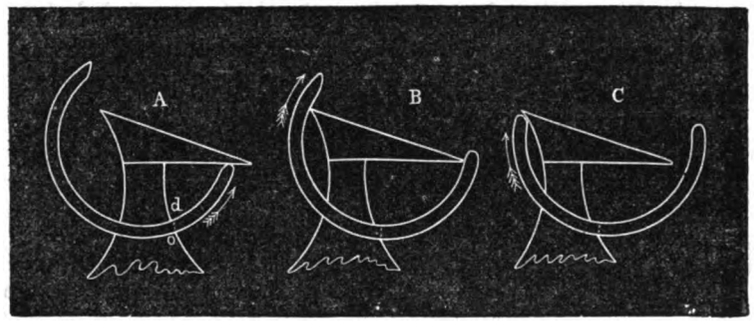

Fig. 29.

Should the pin not be in its proper position, one or even several of the following faults may be met with: (1) over-banking, especially against the small lip, when the heels of the teeth have been cut to a rather sharp angle; (2) striking of the end of the slot against the U-arms; (3) frequent striking of the banking pin against its stud. The pin must be moved slightly to the right or left according to the side on which the fault is observed.

The method adopted in practice for ascertaining whether the pin and stud are correctly placed is as follows:

The spring having been wound up one or two turns, the balance is checked by a strip of thin paper, and is caused to travel first in one direction and then in the other by means of a piece of pegwood, until the pin strikes against the banking stud. If the complete rotation can be performed without any of the above faults showing themselves, both are properly located; but if over-banking occurs, if the point of the tooth comes too near to the edge of the lip, or if the wheel recoils slightly when the pin comes near to the stud, it is essential that the position of either the pin or stud be altered.

When the pin is resting against the stud there must always be a slight play between the U-arms and the end of the banking slot; any strain due to a shake must be supported entirely by this banking pin.

426.—With a ruby cylinder there are two studs. Anyone who has thoroughly mastered the subject up to this point will be able, without difficulty, to ascertain whether these two studs, which limit the extent of the vibration, are essential; and whether by bringing them closer together, or by moving the pin slightly to the right or left, it will not be possible to increase the arcs of oscillation and, by so doing, to secure more perfect timing.

To mark the lifting points and the position of the balance-spring stud.

427.—Let it be required to mark on the plate of a watch two points to indicate an apparent lift of 40°.

Measure on the *Grammaire* (see paragraph **507**) a circular arc of 40° by means of a pinion gauge; that is, rather more than one-third of the diameter of the balance. Then mark on the plate below the balance, and corresponding to the two points of the gauge, two *lifting points* (*a* and *n*, fig. 32, page 239), so named because they indicate the extent of the total lift. A third point *c*, called the *central point*, must be made exactly midway between *a* and *n*.*

One or two turns having been given to the key, cause the lifting action to take place, observing it very carefully with an eye-glass, and at the same time moving the balance, which is held by a piece of paper, by means of a pegwood stick. Immediately on a drop occurring keep the balance stationary, and make a mark with rouge on its edge opposite to the first lifting point (that is, the last reckoning in the direction in which the balance is moving). The cylinder is now gently brought backwards until the second drop occurs, and another mark is made opposite to the other lifting point. If the two marks coincide in one and the same spot, it indicates the position for the reference dot on the rim of the balance; but if they are separate this dot should be placed midway between them.

The balance is next brought backwards until the dot is

* These marks should be *below* the edge of the balance, because if they were outside it they would no longer give the lifting arc as measured at this rim of the balance, but an arc which is the smaller according as they are more distant from the balance, and would, therefore, give an erroneous measure of the lift.

over the central mark, and the balance-spring stud should be examined to ascertain whether it lies in the same direction. If it is to the right or left, a little rouge should be placed on the balance directly beneath it; the balance is then carefully removed and the reference dot is made on its edge, a small mark being also made on the top of the rim to indicate the position of the stud, and thus facilitate the replacing of the balance-spring whenever it has been removed.

It may be observed that usually it is only necessary to make one mark on the balance, if care has been taken when placing the lifting marks on the plate to make the central point come accurately below the middle of the ear of the cock; in other words, of the stud itself.

Short Method of Verifying a Horizontal Escapement.

428.—This verification can only be efficiently performed on a depthing tool, provided with a graduated sector as is explained subsequently in the article on escapement planting; but in ordinary watch-jobbing, and even in examining, it is possible to verify the escapement with sufficient accuracy before taking the watch to pieces, and therefore, of course, without resort to the use of the tool.

It is first necessary to ascertain that the reference dot on the balance is properly placed, that the lifting points on the plate (*a* and *n*, fig. 32, page 239) are at the correct distance apart, and that the central point *c* is accurately midway between them.

About a quarter of the arc *a c* or *c n* is now marked off on either side of *c*. These arcs, 1 *c*, *c* 2, to the right and left of *c* measure about 5° each.

The spring having been wound up one or two turns and the balance held by paper, it is caused to slowly rotate by a pegwood point, and the several functions are tested in the manner described in the next article.

If the result of this examination is satisfactory it only remains to clean the escapement and put it together again, after having put the remainder of the watch in thorough repair.

When any faults are ascertained to exist in the escapement they must be corrected as explained in the following paragraph, but it must always be remembered that the empirical data there given are not to be taken as in any sense invariable, and a watchmaker who is thoroughly cognizant with the theoretical

considerations and experimental data explained and set forth in the preceding chapters never need be at a loss as to the means to be adopted for securing, with any given cylinder escapement, the most perfect rate that its quality admits of.

Planting the Escapement.

Verification on the Depthing Tool.

429.—It has been shown that when the inclination of the impulse planes is insufficient the *real lift* is not of the requisite extent; for if, in such a case, we deepen the escapement with a view to increase the lift, a setting, increased friction and excessive external drops must result (**413**).

The reader has also seen that when the inclination of the teeth is excessive, it becomes necessary to increase the motive force in order to avoid setting, and then the wear of the edges becomes more rapid, and the banking of the balance more frequent; if, in order to avoid these effects, we make the escapement shallower and so reduce the lift, we give rise to excessive external drops.

In a previous article (**406**) we have enumerated the errors that are occasioned by employing cylinders either too much opened or too much closed. In another article (**409**), the characteristics of a cylinder that is too small or too large as compared with its escape-wheel are given.

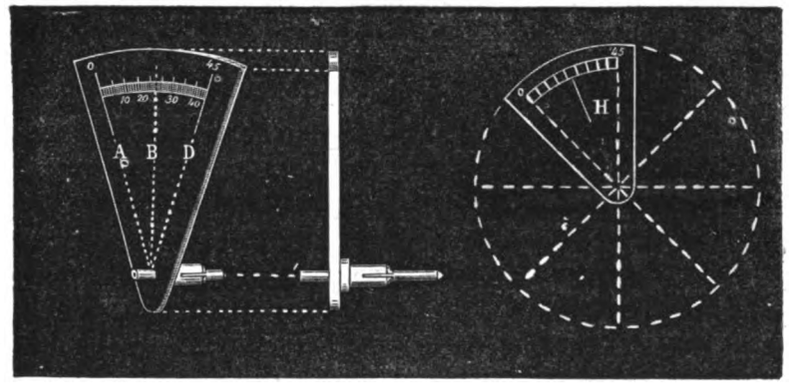

Fig. 30.

Bearing these facts in mind, take the cylinder and the wheel fixed on its pinion, or on an arbor, and adjust the two in the depthing tool, having previously attached the appliance represented at A B D, fig. 30, to the centre which carries the cylinder.

This appendage, as will be seen, consists of a triangular

sheet of brass riveted on to a collar of the same metal. This collar is split throughout the greater part of its length so as to impart elasticity to it, and thus to hold the brass sector in any desired position on the centre.

The lines A and D enclose an angle of 45°, which is divided on a circular arc into degrees, or at any rate into intervals of 5°. The line B must accurately bisect the angle and may be called the *central line.**

If the cylinder is by itself, it must be provided with an index by means of which the angle of lift can be observed. If the balance is attached, the dot on its rim or a small rouge mark will serve as an index (**461**).

Assume that a medium size lady's watch, which should show an apparent lift of 45°, is under examination. We have selected this number because, as has already been pointed out at the end of paragraph **331**, such watches are frequently met with in practice.

The two arms of the tool are brought gradually together and the wheel is caused to rotate by the finger, occasioning an oscillatory movement of the balance. The appendage A B D is turned on the centre until the dot on the balance rim or the index attached to the cylinder performs half of this oscillatory motion on either side of the central line.

The wheel should fall to rest before occasioning a total lift of 35° or 38°, and this lift should be tested up to about 50°, in order to make sure that it will be possible, if necessary, to deepen the escapement; then bring back the middle of the straight incline to coincide with the centre of the cylinder (**430**).

The lifting action must now be gently effected by means of the fingers, one pressing upon the wheel lightly and another guiding the balance, and the movement of the wheel is carefully watched. If the tooth begins to lift when the mark on the balance is about 3° short of the central line B, the escapement is satisfactory, and it only remains to trace it out as in the case

* If a dividing engine is not available for this purpose, divide a round plate, perforated in the centre, into eight equal parts, each of which will contain 45°. Then subdivide one of these parts into 9 portions, each of which will represent 5°. Draw the central line and carefully cut out the triangle so divided; it will be very approximately correct if the operation has been performed with care (H, fig. 30). The protractor supplied in a box of mathematical instruments can be employed for this purpose.

of an ordinary depth, placing one point of the compass in the pivot-hole of the escape-wheel.

If instead of three degrees there are four or five, it indicates that the cylinder is a little too open, or the inclination of the impulse plane somewhat excessive. Such faults may as a rule be overlooked; but if the difference exceeds five degrees, the opening of the cylinder must be verified on a Jacot or cylinder gauge. If the cylinder is found to be too open it must be replaced. If the opening is correct, the error must be sought in the teeth, which are too much sloped. They must be re-adjusted on the incline tool, removing as little metal as possible, for the operation has the effect of increasing the drop.

When the wheel is observed not to commence its lifting action until the mark on the balance has reached the central line or even gone beyond it, this indicates that the cylinder is too much closed or the tooth not sufficiently inclined. The opening of the cylinder must be examined into and corrected, if necessary, by reducing the entrance lip by means of an iron file and rouge or a ruby file. But if it be found correct the fault lies in an insufficient inclination of the teeth. They must be rendered more sloping, and if the workman is not provided with a special tool for this purpose he must reduce the tooth at its point, gently rounding it, and thus incline the point towards the centre of the wheel; he must then polish the inclines lengthwise, first with rouge and finishing off with a burnisher.

It is advisable to practice the verification of escapements in the depthing tool until their principles have been thoroughly mastered; afterwards, when repairing watches, it will suffice to verify them in the manner explained in paragraph **428.**

The same method is resorted to in determining the position or verifying the proportions of all these escapements, with this difference, that, if the total lift exceeds or falls short of 45°, the half-lift ought to commence from 1° to 2°, or even 3° further from the central line in the first case and nearer to it in the second.

To cause the middle of the incline to pass through the centre of the cylinder.

430.—The only accurate method of setting the escapement at its *true point* consists in causing the middle of the incline to pass through the centre of the cylinder (**413**).

In practice one is frequently compelled to depart from this principle in order to render serviceable escapements that are not accurately proportioned; but it is useful, as a starting point, to

first set the impulse plane in the exact position that it would occupy if all the parts were constructed in accordance with established rules.

When the heel of the tooth escapes from the engaging lip, and the point falls on the internal locking surface (at *c*, fig. 31), if we assume that the middle of the straight incline coincides with the centre of the cylinder, it is evident that, as this incline represents a diameter, the entire amount by which the half-shell exceeds 180° must project beyond the point of the tooth. This projection will be 16° in the case of a half-shell of 196°. When we remember that the rounding of the great lip occasions a slight loss on the lift, and that the incline of the disengaging lip occupies 10° of circumference, it is evident that, in practice, the point of the tooth cannot fall on the locking surface at a distance of more than about 4° from the beginning of the incline of the small lip.

The external locking will also commence at a distance of about 16° from the engaging lip; but, since the very pronounced inclination of the small lip occasions a loss of several degrees on the lift, and further, since, in consequence of this inclination, the tooth only moves the external surface of the cylinder during the second half-lift through the same actual distance as that through which the internal surface is displaced during the first half-lift, so that the arcs have not the same value when expressed in degrees, it follows that the tooth will fall on to the external locking surface about 12° from the edge of the great lip.

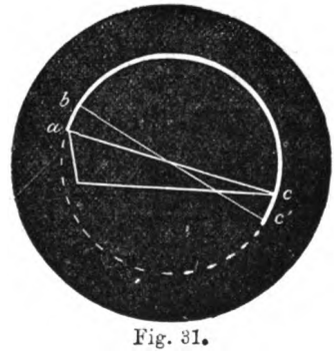

Fig. 31.

A change in the height of the incline will not occasion any variation in the distances between the cylinder lips and the points on which the tooth falls.

For let *b c'* (fig. 31) be the more elevated incline, its point falls at *c'*; but this is precisely the same as that occupied by *c*,

for the cylinder, being moved to *a* by the plane *a c*, is moved up to *b* by the plane *b c'*.

If the half-shell exceeds or falls short of 196°, it will be necessary to add the difference in the first case, and subtract it in the second to or from the amounts above given.

This being granted, the following is the method by which we can practically ascertain whether the middle of the straight incline is placed centrally.

Having previously taken the measure of the half-shell, the balance is placed in position and, while checked in its motion by a piece of paper, is caused to rotate until, with a half-lift of 20°, the point of a tooth has reached the inner corner of the engaging lip (rather more in the case of low inclines and less if the inclination is considerable). In this position of the escapement, if the middle of the straight plane is central, it necessarily follows that the heel of the tooth on the other side of the cylinder must be on the prolongation of the diameter, and consequently the whole portion of the half-shell in excess of 180° will project beyond the heel. Thus with a half-shell measuring 196°, this amount will be 16°, if 200° it will amount to 20°, etc.

By pressing the centre wheel backwards so as to bring this heel against the cylinder, and then moving the balance in the reverse direction until the heel escapes from the small lip and falls within the cylinder, it is easy to ascertain whether the projection is sufficient.

If the opening of the cylinder be determined with care, if on examination the form of the lips is found to rigidly conform to the conditions already laid down, and if the above circumstances be borne in mind, it will always be possible, without much difficulty, to make the middle of the plane coincide with the centre of the cylinder very approximately, or at any rate to occupy the position which secures relatively the best going of the mechanism (**414**).

Note on Timing in Position.

431.—The difficulty experienced in maintaining the rate with a change from the horizontal to the vertical position is due to the difference in the total friction in the two cases. It will be evident that any inequality in the size of the pivots, their shape, the thicknesses of the holes, and the extent to which they are cupped, etc., will vary the ratio between these totals.

Generally speaking the thickness of a ruby is equal to the diameter of its hole, and the ends of the pivots are made perfectly flat.

When the foregoing operations do not suffice to secure a satisfactory rate in the two positions, some watchmakers set the balance out of equipoise, proceeding as follows:—

The watch having been accurately timed when laid down, is suspended.

If it *gains* in this position a small quantity of matter is removed from underneath the *bottom* of the balance. The upper portion of the balance being then slightly heavier than the lower, the arcs described will be rather longer and slower in this position without sensibly affecting those in the horizontal position, and the requisite retardation will thus be secured when the watch is suspended without changing its action in a horizontal plane.*

If, on the other hand, the watch loses when suspended, the matter must be removed from the *top* of the balance, and then, since the bottom will be slightly the heavier, the arcs will be somewhat shorter, and a gain will occur in this position.

A balance must not be put out of equipoise unless it be found absolutely necessary; and in any case it must be done with very great care and to a very slight extent, frequently testing the effect produced lest the requisite amount be exceeded.

This operation, which has long been known, for it is referred to in the works of Berthoud, can only be resorted to when the oscillation does not extend beyond a complete circle. A greater oscillation would occasion effects directly the opposite of those which occur with arcs less than 360°. Besides, though it may sometimes be advantageously practised with common watches, it is not to be depended on for those that are carried in the waistcoat pocket and there take up variable positions; it must, therefore, not be resorted to with watches from which real precision is expected.

In fact, if a watch be constructed with care, if the escapement be made in accordance with the principles we have already laid down, if the balance is perfectly equipoised and not too heavy, the timing in position will be so nearly correct as to render any interference with the equipoise of the balance un-

* The *top* and *bottom* of a balance are determined by a vertical line assumed to pass through the centre of the balance when the watch is suspended by the bow.

necessary; for it must be always remembered that the sole obstacle to this timing consists in the difference between the total amounts of friction in the two cases, and that it is always possible to equalize these two amounts, as will be seen in the article in the third portion of this work which is specially devoted to the consideration of this important subject (see paragraphs **423, 1438, 1442, 1450**).

To Rapidly Time a Cylinder Watch.

Determination of the number of Vibrations.

432.—Several methods may be resorted to when it is required to promptly regulate a watch, and we proceed to indicate a few of them.

It is essential, in the first instance, to ascertain the number of vibrations that should be performed by the balance per hour.

When the watch is taken to pieces for repair, the teeth of the wheels and the leaves of pinions are counted, and the number of oscillations that correspond to the several dimensions is ascertained from tables prepared for the purpose (see the chapter on the *Calculation of Vibrations*, Articles **1056** to **1060**).

This simple preliminary operation occupies barely three or four minutes if the requirements of these tables are properly complied with.

If it is not desired to completely take the watch to pieces, the number of vibrations can be estimated by the aid of *Timing Balances* (**439**), or by the time occupied by two wheels in performing a rotation. Such a determination is usually very easy, for, in most modern horizontal watches, the fourth wheel describes 60 revolutions for each one of the centre wheel. This fourth wheel therefore rotates exactly once per minute, and the balance of such a watch makes 18,000 vibrations per hour; that is, 300 per minute.

Having ascertained the number of vibrations, the desired end may be attained by either of two methods; one of which depends on the ear, and in the other comparison by the eye is resorted to.

To Regulate by Counting the Vibrations.

433.—We will take the most frequent case, in which there are 18,000 beats per hour.

The springer, knowing that 300 impacts occur in a minute, or 150, if, as is more convenient, only one be counted for every

two strokes, fixes his eyes on the seconds hand of the regulator, at the same time holding the watch to his ear. When the hand stands at zero he begins counting the vibrations, and continues (either uninterruptedly, which is somewhat difficult, or in series of 10, 20, or 50 blows) until the hand has completed a revolution.

If the number of impacts thus counted is exactly 300, or 150 when every two are counted, the watch is regulated. The counting may be repeated for security.

Should the number fall short of this amount the index must be moved towards *fast;* in the contrary case it must be set back and the counting repeated. This process is continued until the adjustment is found to be satisfactory.

In a few minutes, unless the construction be at fault, the watch will be so far timed that the watchmaker can part with it almost immediately.

434.—Or another method may be adopted. This consists in employing a second watch, accurately timed, and having the same number of vibrations per hour. With a little practice it is easy, by holding a watch to each ear, to ascertain when the vibrations are in exact accord (see the *Handbook*).

The Vibration Counter.

435.—Considerable skill, which however can always be acquired, is necessary in order to count three series of 50 each without error; but by employing the *Vibration Counter* of M. Leclerre, described in the *Watchmakers' Handbook*, this operation is materially facilitated even though the counting be carried on to 300.

It is then only necessary to count from 1 to 10, and each 10 is registered by the instrument. This consists of a ratchet wheel with 60 teeth, which is maintained in any position by a click, and acted on by a pin provided with an external button. This pin carries at its inner extremity a hinged catch or pawl to engage with the teeth, and thus cause the wheel to advance one division each time the button is depressed by the finger; the pin resumes its position as soon as the finger is removed, through the action of a spring.

The axis of the wheel carries a hand which traverses a scale of 60 divisions.

Keeping the eye fixed on the regulator the workman commences to count, as explained in paragraph **433**, holding the

watch against the ear with one hand, and having the counter in the other.

Each time the word *ten* is uttered the button must be depressed, care being taken to immediately remove the finger for the pin to resume its position. This is repeated until the seconds hand of the regulator has performed a complete rotation. If, when it stands over the 60, the word *five*, for example, is uttered in counting, this figure is noted, and the number of tens registered by the counter is ascertained. Suppose there to be 29 tens, or 290 vibrations; adding 5 of the last incomplete ten we have 295 vibrations per minute. The balance-spring is then too weak, since there is a loss of *five* vibrations per minute.

If, on the other hand, the counter indicate 30 or 31, we should have in one case 305, and in the other 315 vibrations in a minute. The watch would, therefore, gain either 5 or 15 vibrations in each minute.

To Time rapidly by Comparison.

436.—In order to employ this method it is necessary to be provided with a watch-movement accurately regulated, and beating the same number of vibrations as the watch to be timed.

The two movements are placed horizontally in the same plane, in such a manner that the balances are in close proximity, and, when at rest, two of their arms are in a straight line; this arrangement is indicated in fig. 32.

The two balances are now set in motion, either by momentarily touching the two arms P and H, or by bringing the two balances in contact and suddenly releasing them, or by any other suitable means. (See the following article.) It is not necessary that the extent of the vibrations be considerable. A very little practice will enable the workman to set the two in motion simultaneously, and any error is at once detected by the two arms failing to pass the line of centres, *b d*, at the same instant.

The balances must now be carefully watched; if the arm H always passes this line at the same time as P, the watch is regulated, and the experiment need only be repeated once or twice in order to make certain. If H (that is, the watch under adjustment) gains on P it will be necessary to move the index of H towards *slow* and towards *fast* in the converse case.

By repeated trials of this nature the watchmaker will

soon attain to considerable skill in comparing the movements of the two balances and in detecting any slight discordance.

437.—*Observation.*—The two watches should rest on a firm support of considerable mass, and the period during which they continue in action should not be much prolonged if the main-springs are wound up. Every minute or two they should be stopped at the resting point of the balance-springs in order to avoid slight errors that are occasioned by the mutual action of the two reciprocating bodies.

To measure the strength of a Balance-Spring.

438.—We shall give in the *Handbook* the description of a tool to measure the strength of a balance-spring; but the following process, which is only a modification of the methods above described, will suffice in all the ordinary work of the watch-jobber.

He must be provided with a *Timing Balance* accurately timed, taken from a disused watch beating 18,000 vibrations, the whole mounted on a plate in the manner in which P is mounted on the plate A C D B. After having ascertained that the timing balance is provided with the requisite supply of oil and vibrates with perfect freedom, the workman proceeds as follows:

The new balance-spring intended for the watch under repair is attached to its balance by the collet; this balance is

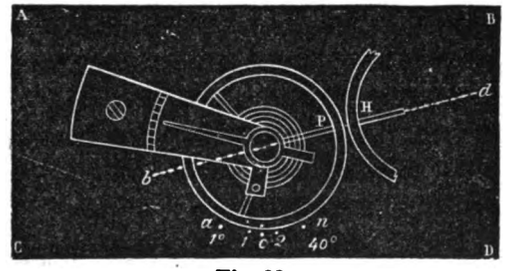

Fig. 32.

raised by pinching in the tweezers the external coil of the spring and the point of the lower pivot is allowed to rest lightly on the plate A C D B, an arm of each balance being set in one straight line as already explained. Maintaining the detached balance in this position the two are set in motion in the manner described in paragraph **436.**

With the reference balance somewhat heavier than the other it is easy to start their vibrations in unison by

momentarily allowing them to remain in slight contact and then separating them.

The point at which the spring is held in the tweezers must be moved inwards or outwards according as its action is too weak or too strong, as the length of the vibrating strip is thereby altered; the operation must be repeated until synchronism of the two oscillations is secured.

The point held in the tweezers indicates the position at which the spring should be pinned in the stud; for the index of P is not provided with curb pins on either side of the outer coil of the balance-spring. By this method, then, it is possible to ascertain very rapidly at what point the requisite resistance is offered by the spring; but when in position in the watch, it still remains to be determined where the index should be placed, and this must be done by one of the two methods subsequently explained.

Observation.—If the two pins are very close together, so that the spring is practically gripped by them, the point held by the tweezers should come between them; if they are very far apart, so that the balance-spring barely touches them when pushed towards *slow*, this point will, as already mentioned, be pinned in the stud. Enough has, however, been said on this subject to ensure that the intelligent watchmaker will be able to judge, very closely, at what distance from the stud the point held by the tweezers should be fixed.

In springing watches many workmen in factories resort to the method here explained. It enables them to select a spring and pin it in position almost with a certainty of being correct and materially shortens the time occupied in timing.*

Timing or Comparison Balance.

439.—It would be very useful if every watchmaker would provide himself with several balances mounted on plates and having balance-springs attached which give different numbers of beats, say 4, 5, 6, etc., per second. The number of vibrations in an hour should be engraved on each plate (**1453**).

Such appliances would be serviceable not only for ascertaining the strength of balance-springs, but also for determining the number of teeth or leaves of a lost wheel or pinion. When

* A still more simple method of ascertaining the strength of a spring, held in the tweezers as above, consists in counting the vibrations of the balance during a minute with the aid of a regulator or watch with a seconds dial.

we published this method we were the first to indicate this second convenient application which is explained in detail in the article headed *To replace a lost Wheel* (**1180**).

Note on a Novel Means of determining the Weight of the Balance.

Half-Timing.

440.—A body, that is oscillating under the influence of a given motive force, requires a very short time to attain its maximum arc of vibration when it is light, and a proportionately longer period as its weight is increased (**323**).

It is a consequence of this physical law that when the balance of a watch is too light as compared with the motive force, it at once attains to its maximum oscillation, while if it be too heavy the first vibrations are very short, and they gradually increase in extent until the normal arc is performed.

In some watches it is not until the fifteenth oscillation that uniformity is observed.

We cannot here discuss the method in detail as our experiments on it are still in progress. They will enable us to compile a table giving without trial, and for all sizes of watches, the correct proportion between the weight of the balance and the motive force. We would at once beg our brother watchmakers to make experiments on thoroughly good going watches, and to communicate their results to us. A comparison of such results among themselves cannot fail to lead to the enunciation of a practical, safe, and accurate rule, based on an experimental physical law.

The following method should be pursued:

The mainspring having been let down, a rouge mark is made on the plate opposite, say, the banking pin. This mark enables us to set the balance so that the spring occupies its resting position.

The mainspring is now fully wound up, and when the watch has gone long enough to overcome the excessive resistance that opposes the commencement of its motion, the balance is gently stopped in the resting position of the balance-spring; it is then set free, the gradual increase of the oscillations being carefully watched through an eye-glass fixed in a convenient position, and the vibrations are counted until the full amplitude of oscillation is attained.

CALLIPER AND PROPORTIONS OF A CYLINDER WATCH THAT HAS GONE VERY SATISFACTORILY.

441.—In concluding this chapter we cannot do better than lay before the reader the calliper and exact dimensions of a horizontal watch that has gone remarkably well for about *thirty years*.

It bears the name of Robin, Paris. It has all the appearance of high class Geneva workmanship, and its thickness is somewhat less than that usually adopted by makers at the present day.

The several *plays*, or intervals of safety, have been carefully and intelligently regulated. The entire watch is still in a perfect state of preservation. Always after cleaning it has at once maintained its original uniform rate.

We shall regard it as a standard of comparison; it is a substantial confirmation of the statement we are unfortunately so often compelled to repeat; good watches are the result of an intelligent application of the laws of mechanics. From this it must be understood that the same end may be attained by different means, provided only that they are in conformity with those laws.

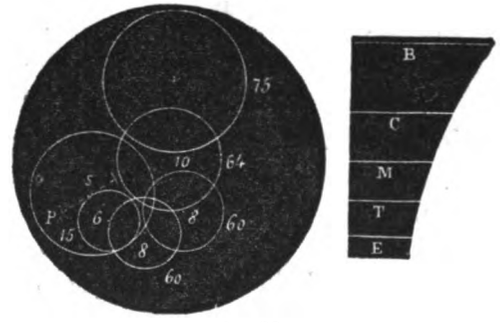

Fig. 33.

The calliper of the watch in question is shown in fig. 33.

On the right are given in a separate diagram the diameters of the several wheels, their distances apart being equal to the radii of the same; B, the barrel; C, the centre wheel; M, the third wheel; T, the fourth wheel; E, the escape-wheel: this arrangement makes evident at a glance the gradual decrease in the motive force.

P is the fixed point of the balance-spring, whose diameter is indicated by the dotted line *s*.

The following are the dimensions of the different parts accurately measured in millimetres and decimals.

Fixed Pieces.

Total thickness of the movement . . . . 3·15 Millimetres.
Plate, diameter 40 mm., thickness . . . . 1·53 ,,
Thickness of the bars . . . . . 1·62 ,,

Train.

The numbers of the teeth of wheels and leaves of pinions are indicated in fig. 33.

	Diameter.	Diameter of Pinions.	Size of Pivots.
Barrel . . .	19·00 mm.		
Barrel Cover . . .	17·25 ,,		
Centre Wheel . . .	13·50 ,,	2·60 mm.	0·93 mm.
Third Wheel . . .	10·70 ,,	1·70 ,,	0·20 ,,
Fourth Wheel . . .	9·96 ,,	1·45 ,,	0·15 ,,
Escape-wheel . . .	8·25 ,,	1·05 ,,	0·10 ,,

Motive Force.

Barrel.—Internal diameter, 16 mm.; diameter of the barrel arbor, 5·2 mm.

Mainspring.—Width 1·1 mm.; thickness, 0·2 mm.; length, 59 centimetres.

This spring was wound up half a turn for setting the stop-finger; it maintained equilibrium with the following weights (placed at the end of a wooden lever 185 millimetres long, and weighing 3·88 grammes) during the winding up of the watch:—

At the first turn of the key . . . 6·0 grammes.
At two-and-a-half turns of the key 7·5 ,,
At the fourth turn . . . . . 9·0 ,,

In going for twenty-four hours, the energy of the motive force fell off to the extent of very nearly one-third of its full amount.

Escapement.

Balance.—Diameter 16·50 mm.; weight (estimated from that of a broken balance of the same dimensions), 0·14 grammes.

This weight was thus distributed: In the rim . . 12 } 14
In the arms and centre . . 2 }

Cylinder.—Diameter . . . . . 0·95 mm.
Opening (measure of the half-shell) . . 0·55 ,,
Size of pivots . . . . . 0·11 ,,

Their length is two-and-a-half times their diameter. The pivots of the escape-wheel are a shade finer than those of the cylinder.

Balance-Spring.—Total length, 182 to 183 mm. Diameter, 8 mm., which comprises $11\frac{1}{2}$ coils. The hole in the stud is 4·50 mm. from the centre. Diameter of collet, 1·85 mm.

Lift.—Apparent lift 36° (the incline is slightly curved).

Lift on one side . . 26°.

The *real* lift is therefore about 52°.

There are four jewels to the escapement, and all the pivots of the train work in brass.

As will be gathered from the numbers given in figure 33, the watch makes 18,000 vibrations per hour.

Half-Timing.—When the mainspring is wound up one complete turn above the point at which the stop-finger holds it, the normal extent of vibration is reached between the 20th and 22nd oscillation.

When fully wound up it is attained between the 14th and 16th oscillation.

The extent of the arc was found to alter slightly when the mainspring, from being only wound up one turn, was made to exert its full power.

Ratios and Sundry Observations.

442.—The lines B, C, etc., in figure 33, graphically represent the manner in which the motive force decreases, or, rather, the levers that transmit it.

The action of the escapement commences with the first turn of the key, so the force is evidently considerable, and yet, thanks to the excellence of the proportions adopted, the balance does not in any way suffer from this excess of force; a fact which the two following circumstances go to prove:

(1) It performs 15 vibrations before attaining to its normal amplitude when the mainspring is fully wound up.

(2) This amplitude varied but slightly with a change of nearly one-third in the amount of the motive force.

The effect of this force is well proportioned, and it cannot interfere with the timing since it decreases with uniformity; this is shown by the proportion between the several weights which it maintains in equilibrium when the tension is varied. It will be understood that this result is in part influenced by the diameter of the barrel arbor and by the length of spring that remains always in the same position at the ends, as these portions are without effect on the going of the watch; that is to say, they lie beyond the limited range of the four revolutions allowed by the stop-work.

Representing the balance by B, the cylinder by C, the opening

by o (the diameter of c being called 100), and the pivots by p, the several proportions were found to be as follows:

B : C : : 17·3 : 1.
C : O : : 100 : 57·8
C : P : : 8 : 1.

The proportion which the balance bears to the cylinder is excellent, as will be seen from the experimental data given in article **388**. The opening of the cylinder gives a half-shell of between 196° and 200°; and the size of the pivots is that usually adopted by experienced practical men.

It is thus seen that the prolonged good going of this watch need not in any way cause surprise, when the many elements of success that combine in it are taken into consideration.

Other proportions might be deduced from the figures given above, but, before conclusions of any value can be deduced from them, it is essential that they be compared with those observed in watches of other callipers. We, therefore, trust that experienced watchmakers will communicate to us results analogous to those described in these articles; they shall be published along with our own observations. At the same time we must remark that it is not worth while thus studying a watch unless it has gone uninterruptedly for ten years at least without being meddled with by watch-jobbers.

CHAPTER VI.

CAUSES OF STOPPAGE AND VARIATION IN THE CYLINDER ESCAPEMENT.

Wear or Pitting of the Cylinder.

443.—Wear of the cylinder is usually occasioned:

(1) By the presence of a kind of gritty mass formed by the mixture of oil and dust, or due to the fact of the oil being dried up. The friction then occurs between dry surfaces;

(2) Imperfect polish. The rubbing surfaces not being in sufficiently close contact, the roughnesses of the metals interlock and wear away; this increases the friction, and the form of the parts is modified;

(3) Bad quality of the metal, which, from being brittle and of varying texture, has not a uniform hardness throughout; the polish is bad or unequal.

(4) Bad hardening, the steel being distorted if over-

heated or too soft when the temperature is not sufficiently elevated;

(5) By the objectionable practice of certain escapement makers, who, in order to facilitate their work, temper the lips and locking surfaces of the cylinders to a *straw-violet*, or even a *blue* colour.

(6) A wheel that is not hardened or has been tempered too much.

Such faults will be avoided if care is taken not to allow the escapement to go too long without oil, to always employ the very best steel in its construction, to harden with the greatest possible care, and to abstain from subsequently tempering the lips or locking surfaces, only tempering the teeth to a yellow, and, lastly, to slightly round, in a beaded form, the impulse planes and their points, polishing very highly all the rubbing surfaces.

To re-polish the cylinder.

444.—When a cylinder is but slightly worn, it will suffice to simply re-form the lips, and for this purpose it is well to be provided with two small ruby files, one somewhat coarse, and the other fine. These lips must be very highly polished, and the impulse planes against which they have acted should also be polished. (Whenever the opening is too great the cylinder must be replaced. In such a case see Chapter VII., article **472**, *To pivot a cylinder.*)

Mechanical means may be adopted for polishing the lips. We shall subsequently describe an apparatus for this purpose.

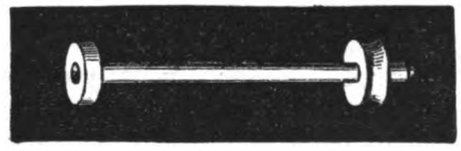

Fig. 34.

It is usual when re-forming the lips to hold the balance between the fingers of the left hand. A more convenient method, however, when operating on the engaging lip, consists in setting the balance in wax on a plate at the end of a hollow arbor, similar to that represented in fig. 34, except that the plate is thinner, and the ferrule removed.

In the case of the smaller lip the balance is fixed with wax by its under surface to the extremity of a rod perforated with a hole rather larger than the cylinder, which has been previously

filed away to half its thickness; by such an arrangement the cavity containing the cylinder is made accessible.

Such an arbor can, as in the former case, be provided with a thin plate; the perforation is large enough to admit the collet.

With both of these arrangements a pair of plates may be employed, one being attached to the other by three small screws. The balance is then clamped between the two, and the use of wax rendered unnecessary.

To re-dress a bent Cylinder.

445.—When a cylinder is bent this usually takes place at the pillar. Several methods may be adopted for correcting such an error.

1. Remove the lower plug and fit a new one. The point of the axis of this latter is filed until the locking surfaces of the cylinder and the rim of the balance run true. Then the pivot must be cut on this axis, the eccentricity of the smaller shell being ignored.

2. If the pivots are broken and the cylinder strained, both plugs must be removed. A perfectly smooth cylindrical arbor which fits the undamaged portion of the cylinder is obtained, and the shell enclosed in a brass screw-ferrule that binds over the entire external locking surface; the pillar is placed in contact with a heated rod. On its assuming a bluish tint, the arbor is carefully pushed into the cylinder, the lamp removed, and the cylinder held by a pair of strong narrow-nosed pliers, one jaw being against the arbor in the banking slot, and the other pressing the middle of the pillar; the whole is then agitated in water until quite cold.

The plugs must now be re-made (**467**).

A strained cylinder in which the pivots are intact can be restored to its proper form by mechanical means.

Setting.

446.—This may arise from:

(1) A deficiency in the force applied to the escapement, whether due to some error in its construction, bad depths, friction occurring between parts that should be free, weak mainspring, etc.;

(2) Inclines that are too elevated (especially if they be straight), to the escapement being set too deep, or to inclines that are too slight butting against the locking surfaces instead of falling on the lips;

(3) Balance pivots that are bent, or have not sufficient play in their jewels;

(4) Badly polished cylinder, or a similar fault in the inclines or points of the teeth;

(5) A balance that is too heavy as compared with the motive force;

(6) A feather-edge or burr left at the heel of the teeth;

(7) Lastly, to the tooth being sometimes held within the cylinder unable to escape because the point is too sharp and badly polished and so catches against the inner corner of the incline of the small lip, which has not been sufficiently rounded, in order, as it were, to make it continuous with the internal surface of the shell (**407**).

We have not included among the causes of setting those which arise from the thickening of the oil and the wear of the cylinder edges.

Any watchmaker will be able, with a little care, to discover the causes of a setting, and to find the best means of correcting them, whether several combine, or only one has to be considered.

An Escape-Wheel out of Upright.

447.—If it leans forward or is too low in the slot it will rub against the lower plug and sometimes even against the fourth wheel.

If leaning backwards or too high in the banking slot, the under face of the disengaging lip will rub against the U-arm, causing the wheel to recoil; and it may even happen that friction occurs between the escape-wheel and the seconds or fourth wheel.

Such a fault must be corrected by setting the wheel straight by moving the steady pins of the bar, and, if this is liable to derangement on turning the screw, a third steady pin may be inserted when the wheel is upright.

When it is impossible to alter the position of a wheel that is too high or too low as compared with the position of the banking slot, the cylinder itself must be moved so that the flat of the wheel passes centrally in the slot.

Such faults as these may be occasioned by the wheel being carelessly riveted to its pinion, or to its not being of the same thickness throughout.

The latter error is corrected by laying the wheel on cork

and reducing by means of an iron file charged with oilstone-dust.

See wheel that is out of flat, etc., page 255.

Too Thick Wheel.—Strained Wheel.

448.—The total thickness of the wheel may be excessive or only that of its flat.

The former case, although rare, is sometimes met with in thin watches. The wheel must be removed from its pinion and reduced by rubbing on a smooth sheet of glass charged with oilstone-dust.

When the flat only is too thick, the height of the banking slot must be increased by means of a small flat ruby file (**453**).

A wheel that is strained must be restored to shape by the method indicated in Chapter VII., when considering the hardening of the wheel.

Insufficient Hollowing of the Escape-Wheel Passage.

449.—If any friction occurs against the teeth during the motion of the wheel, it will give rise to stoppage and irregularity. When there is not a sufficient interval between the flat of the teeth and the escape-wheel cock, the oil fills it and renders accurate timing impossible.

The dust and minute fibres that are collected by the oil will not occasion a stoppage if the interval be sufficient.

A wheel being out of flat or badly riveted to its pinion, pivot-holes that are too large, or a bar that is bent upwards will occasionally give rise to faults similar to those due to too little play. When a doubt exists as to this play, the question may be set at rest by placing polishing rouge on the flat of the teeth and observing whether it passes without contact.

The layer of rouge should be somewhat thick, for otherwise, were it to spread and form a thin film, it would fail to give a reliable indication.

The wheel is set right either by adjusting the steady pins of the bar, or, when no other means are available, by hammering its under side and thus lengthening it. This is accomplished by the use of a large punch shaped like a hammer-head.

An analogous method will answer for displacing the lower pivot-hole when necessary; but there is some danger of breaking the stone, and discretion must be exercised when performing the process; it must only be resorted to when adjustment by any other means is quite impossible.

When the watchmaker knows how to set jewels (see the articles on this subject in the *Handbook*), it will be a simple matter to remove one that is improperly placed, increase the hole which held it, and to repair it by first inserting firmly a small piece of brass; in this the jewel is carefully reset.

The passage is cut on the lathe, the bar having been first fixed by wax to a chuck, or preferably on the Geneva mandril tool, the head being turned by the palm of the right hand while the slide-rest carrying the cutter is gently advanced by the left until a small quantity of brass has been removed. When the cutter has traversed the entire escape-wheel passage, slowly removing the metal, the slide-rest is brought back to its initial position, and the cutter again caused to remove a thin film of metal, the operation being repeated until the necessary depth is attained; by this means neither the bow nor the handle of the mandril is used.

When the bar is extremely thin, or if from any other cause the hollow cannot be cut away, it will be necessary to bend the bar upwards where the hollow would be cut, the two ends being depressed. But this can only be regarded as a makeshift.

The Drops too Short or Unequal.

450.—When the internal drop is insufficient, the tooth will rub against the cylinder by the point and heel at the same time. With a short outside drop, the cylinder rubs against the point of one tooth, and the heel of that which precedes it. In the first case a setting will result while a tooth is contained within the cylinder; in the second the cylinder will be in a U-space. This doubling of the friction, even although it may not actually cause the watch to stop, will seriously interfere with the timing. It may be avoided by slightly reducing the points of the teeth, but the watchmaker should first ascertain whether it would not be more beneficial to replace the cylinder, which may not be correctly formed, rather than to run the risk of distorting a well-made wheel, merely in order to render an indifferent cylinder serviceable. (Read carefully the article on *Drop*, **410**).

If the drops are unequal, some being too short, permanent irregularity with occasional stoppages will result as in the former case; the rate of the watch can never be maintained and it constantly loses. In order to detect such a fault, a rouge mark must be made on the edge of the balance to better

observe the oscillations; if they are found to be of unequal extent, and only certain of them attain to their maximum arc, it is highly probable that the fault exists.

If in the shorter oscillations the reduction is observed to take place on the stud side (that is, the side of the engaging lip), the error occurs in the inside drops. All the teeth of the wheel, therefore, must be made equal in length to the shortest, a suitable gauge being employed as explained in Article **402**.

When, however, this reduction is on the other side, it is occasioned by the external drops. All the U-spaces must be examined by employing a smooth arbor, in the absence of a special tool, and be made equal to the largest.

Or a more expeditious means is to test each drop, placing a rouge mark against such as have not sufficient play, and then merely correct them (**410**).

Unequal drops may arise from teeth being strained, either in hardening or from any other cause.

It may be accepted as a general rule that when there is a measurable inequality in the extent of successive oscillations it almost always indicates that either the drops are unequal, or some teeth cause a greater lift than others.

Noisy Drop or very little play in the Cylinder.

451.—Such effects are due to a heel that is rough, and so catches on the lip, or an incline of too great a curvature. When the wheel is thin, the heel at times rebounds against the opposite side, especially when the cylinder is badly polished. By polishing the heel, or slightly reducing the incline the noise of the drop can be caused to diminish very appreciably.

Cylinder planted too high or too low.

452.—The sources of stoppage and irregularity will be identical with those met with when the escape-wheel is similarly out of place. They must be corrected by the cock and chariot, causing the cylinder to rise or fall until the flat of the wheel is in the middle of the slot. If, after this adjustment has been made, the balance comes into contact with the escape-wheel cock, its rim must be raised by slightly bending the arms near the centre. If, on the other hand, there is not sufficient play above, the rim must be lowered in a similar way.

Banking Slot too broad or too narrow.

453.—When the slot is too wide or dovetailed inwards, and the wheel below its proper level or of insufficient total

thickness, it often happens that the point of the tooth rubs so very near to the face of the slot during the period of rest that there is some danger of its catching on this under face of the small lip, at the point at which the width of the slot is the greatest. In such a case the motion of the cylinder is arrested as though over-banking had occurred.

It will be necessary to make the width of the slot the same throughout by means of a ruby file; the wheel must then be raised or the cylinder lowered, but no more play should exist between these two mobiles than what is known to be strictly necessary.

When the slot is too narrow the cylinder must be firmly fixed in wax at the extremity of a perforated arbor, analogous to that used in polishing a cylinder (**444**), and the slot is operated upon with a ruby file.

The U-arms touched by the end of the Slot.

454.—When the end of the slot is struck by the U-arms, it occasions a variation in the rate of the watch: the wheel is made to recoil. We have explained in the article on over-banking (**425**) how the existence of this fault can be detected, and have also described the means of correcting it by moving the banking pin.

When the depth of the slot is insufficient two pins are necessary, but it will be preferable to increase the depth: this is an operation of some delicacy, which must be performed with a flat ruby file, the cylinder being first fixed at the end of an arbor as indicated in the last article.

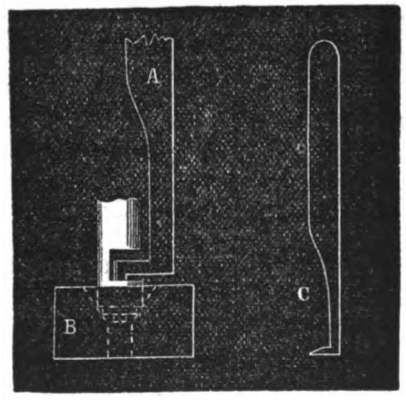

Fig. 35.

A Broken Pivot.—To Replace a Plug.

455.—If a pivot is broken a new plug will be necessary. The old one is removed thus:

A brass block D, perforated with a hole through which the broken end of the axis passes easily, is held in the vice; it is chamfered at the upper end so that, when the cylinder rests in it, it is only supported by the edge of the shell. A hook-shaped punch A (fig. 35), entering the opening of the cylinder, is used to displace the plug slightly, as indicated in the figure.

The projecting portion is inserted into a hole in the riveting stake, which it must fit with but little play so that the shell is supported all round, and the plug driven out with a punch provided with a point of sufficient length.

The plug is then replaced by another that has an axis of the requisite length. (See the articles on *Plugs* and *Plugging* in Chapter VII.

Some workmen, after having firmly fixed the cylinder in wax, drill a hole through the plug, fix a fine steel wire in it, and then turn a pivot; although this method is good in certain cases, it is far preferable to make the new plug of only one piece.

A Loose Plug.

456.—After making a mark on the edge of the cylinder and plug so as to restore them to their initial relative positions, the latter is removed as just explained, and attached by wax to the end of a brass rod, perforated so as to freely admit the axis and pivot. A number of marks are then made over the entire surface of the plug by causing it to roll beneath a good cutting flat file on a hard surface of brass or wood: or marks may be made in all directions with the point of a graver. Some workmen roll the plug in oilstone-dust, pressing it with an iron file, in order to fix some of the powder in the surface of the plug, but good watchmakers disapprove of such a practice, fearing lest particles of the stone that are not held with sufficient firmness between the two surfaces should mix with the oil applied to the cylinder. If, after either of these operations, the plug is still not sufficiently tight, it will be necessary to gently spread the edge of its hollow, employing for this purpose a round hard polished punch that is perforated so as to admit the axis.

Those workmen who cement in the plug with shellac seem to forget that as the small shell has usually very little height, cemented plugs cannot hold firmly in it; besides which the shellac dissolves in spirits of wine.

Over-Banking.

457.—Over-banking occurs when the pin is in the wrong place, or too short or broken. By employing two fine pieces of pegwood, one to control the balance and the other the escape-wheel, and by moving the latter backwards until the tooth is just free of the cylinder, the requisite correction can be made as already explained in the article on over-banking (**425**).

When the points of the teeth have been made to slope downwards, that is to say as shown at s (fig. 28, page 206), banking is not so easy of detection; the cylinder being held less firmly, starts again with a shake, or on opening the case. When the cause of stoppage cannot be ascertained, each tooth should be tried for banking, for it will probably be found that one of the U-arms has been distorted outwards in hardening; there will thus be rather less matter inside the point, in consequence of which over-banking will only occur with this tooth.

A Tooth too Large or too Square at the Heel.

458.—When the teeth are too large or not sloped inwards at the heel, it may happen that the two points *a* and *o* (fig. 36) rub at the same time against the cylinder; very great irregularity will thus be occasioned with an occasional stoppage of the entire mechanism; this may be avoided by sloping the teeth as indicated by the dotted line *m* +.

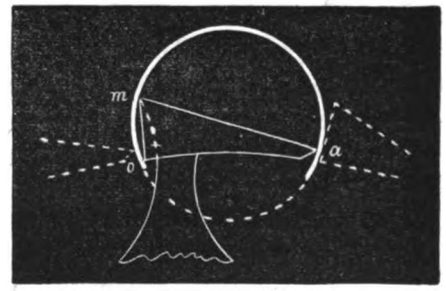

Fig. 36.

If the pillar is situated directly below the point *o*, part of its corner must be removed during the operation.

An Escapement out of Beat.

459.—When an escapement is not in adjustment, or, as it is termed, is *out of beat*, this may be due to any of the following causes: (1) one or even both of the pins that secure the balance spring in the collet and stud are loose: (2) this spring is strained between the two curb pins: (3) the balance-spring stud not having been placed immediately over the dot on

the balance when putting the escapement together. If this mark exists the collet must be rotated until they are superposed. If not, the mark must first be made as explained in the article on reference marks (**427**), and the adjustment then made.

Other Causes of Stoppage and of Irregularities in Timing.

460.—The other causes of stoppage and variation in the cylinder escapement, which any watchmaker can discover and correct with ease, are as follows:—

CYLINDER TOO LARGE OR TOO SMALL (**409**), TOO MUCH OPENED OR CLOSED, ETC. (**406**).

CYLINDER THAT IS OVAL OR NOT PERFECTLY CENTRED. A slight recoil of the wheel is occasionally observed.

A WHEEL THAT IS OUT OF FLAT.—If this fault is not due to the riveting, it may be corrected by placing the wheel on a perfectly true piece of soft metal or hard wood, perforated to receive the pinion; the arm that requires adjusting is then slightly bent by gently striking the upper surface close to the centre. Instead of the sharp end of a hammer-head it is preferable to employ a punch, the end of which is flat with a slight bevel, for this operation.

A WHEEL WITH TEETH THAT ARE TOO MUCH OR TOO LITTLE INCLINED, WHOSE POINTS ARE BADLY POLISHED OR FALL TOO NEAR TO THE EDGE OF THE LIPS, etc.

A WHEEL THAT IS OUT OF TRUTH.—The lifts are unequal. Some teeth fall very near to the edge of the lips, and others at a considerable distance from them.

EXCESSIVE ENDSHAKE IN EITHER THE WHEEL OR CYLINDER. —On inverting the watch, one of these two mobiles will fall through a greater distance than the other, so that their relative positions being altered, there is danger of varying and injurious friction.

ESCAPE-WHEEL PIVOT-HOLES TOO LARGE.—It will have a great amount of play, and all the faults met with when the wheel is out of flat, or not upright, will occur.

A LOOSE COLLET.—The act of opening or closing the case may displace it.

A COLLET THAT IS NOT SUFFICIENTLY CLEAR OF THE COCK.—It will rub either against the screw heads of end-piece, or, if this does not actually occur, the oil between the two will cause variations.

A FOURTH WHEEL WITH TOO MUCH ENDSHAKE.—It rubs against the escape-wheel when the watch is inverted.

A BALANCE THAT IS OUT OF FLAT, NOT UPRIGHT, SET TOO HIGH OR TOO LOW.

BALANCE ARMS THAT TOUCH the stud, curb pins, or escape-wheel cock.

BALANCE-SPRING STRIKING against the centre wheel or the stud; or else its second coil striking the first curb pin.

BALANCE-SPRING RUBBING against the cock, the balance arms, or the centre wheel.

STEEL CURB PINS.—There is danger of rust and magnetization. (See further on: Oil on the curb pins.)

LOOSE CURB PINS.—The watch cannot be timed.

A BANKING PIN THAT TOUCHES the pinion or axis of the fourth wheel, the gong of a repeater, the band of the case when this latter is closed (as then the edge is forced inwards by the pressure of the dome), or the fly-spring; the escapement will start again immediately on opening the case.

A BANKING PIN THAT CATCHES against the face of the stud instead of striking against its side, in consequence of the pin being too short or the face of the stud a little sloped.

A BANKING PIN THAT ADHERES TO THE STUD.—This case although rare in climates such as that of France is much more frequently met with in damper climates, such as England for instance.

RAPPING OF THE BANKING PIN.—This arises from the pin being in the wrong place, the lift too much extended, or the motive force excessive.

It is a fault which often escapes detection for a long time, since it can only occur under certain conditions: for example, when the watch is in wear, or when a hole that was too small has become oval by the wearing away of its side. In some cases the fault is not met with when the spring is fully wound up, but occurs when it occupies a certain position. Under such circumstances it is occasionally possible to correct it by employing a balance-spring that is somewhat shorter with a collet of rather increased diameter.

THE EDGE OF SHELL RUBBING against either the jewel or the setting, owing to a want of space.

JEWELS THAT ARE BADLY SET, that are loose or too thick, so that the pivots do not project externally. Such a fault may of course be due to the pivots being too short.

OIL on the rim of the balance, altering its weight and destroying the equipoise, or else on the curb pins, causing the spring to adhere sometimes during several successive vibrations.

TOO MUCH OIL TO THE UPPER ESCAPE-WHEEL PIVOT.—It is collected gradually by the balance arms.

THE OIL OF THE UPPER PIVOT OF THE FOURTH WHEEL.—It spreads over the axis and thickens. In certain callipers it passes thence to the rim of the balance or the banking pin.

A BALANCE TOO SMALL, OR TOO LARGE, AND OF INSUFFICIENT WEIGHT, CYLINDER TOO LARGE (see size of axes **345**), are very frequent causes, if not of stoppage, at any rate of constant irregularity in the going.

The same effect is observed with a MAINSPRING that has become weak either throughout its entire length, or in parts, in consequence of the bad quality of the steel, or the absence of stop-work.

Ruby Cylinder.

A BODY BADLY CEMENTED, and moving backwards and forwards under the pressure of the inclines.

A BODY THAT PROJECTS TOO MUCH BEYOND THE CORNER OF THE SETTING.—They should be almost on a level, for otherwise it will be necessary to set the banking studs farther apart in order to prevent the corner from striking against the point of the tooth during the locking; the extent of vibration is thus diminished, and the banking pins are more liable to strike against the studs, etc., etc. (see articles **403** and **404**).

Depths.

Bad depths, which cause the motive force applied to the escapement to be very variable, occasion irregularity. Since they are apt to become worse and worse, it may happen that the timing properties of the escapement are in time entirely nullified.

When the escape-wheel pinion is badly-sized, the timing is sometimes indifferent, or only constant for a short period.

For the method adopted in replacing this pinion, see article **502** in Chapter VII.

461.—When speaking of the verification of an escapement on the depthing tool, we mentioned that if there is no balance attached to the cylinder it must be provided with an index or

finger. The reader will understand that this index must be formed of soft brass so as to be easily bent in any direction, and it should be carried by a small screw-ferrule fitting on the collet or upper axis; motion of the escapement may then be produced by pressing gently on the rim of the ferrule.

CHAPTER VII.

ACTUAL CONSTRUCTION OF THE CYLINDER ESCAPEMENT

THE CYLINDER.

462.—The watch material dealers keep an assortment of cylinders of all sizes, but, notwithstanding this fact, every watchmaker will at times find himself obliged to re-make and renew a cylinder. This operation only requires a little care and skill, such as is common in the trade; the workmanship is much less delicate than that involved in the construction of a wheel, with which we shall occupy ourselves in the second part of this chapter (**477**).

After attentively following the details we proceed to give, every watchmaker of ordinary ability ought to be able to make good cylinders, and in a few days to acquire the practical knowledge necessary to do it with some rapidity.

To Measure the Heights.

463.—The heights may be ascertained from the old cylinder, or, in its absence, the following is the best method of procedure:—

A small brass rule is made, about 2 millimetres (0·08 ins.) in width, 2 or 3 centimetres (1 inch) in length, and of the thickness of an ordinary playing card.

Some rouge is then placed on the points of the teeth and the edge of the U-arms. The wheel being in position, the rule is now held vertical, resting on the lower jewel in the position a cylinder would occupy. The wheel is pressed against both sides of it in succession, and the rule is then filed away, as though it were a cylinder, in accordance with the rouge marks left by the wheel. It is again placed in position in order to make sure that the wider slot is of the right height,

and that the U-arm comes midway in the smaller slot. (A, fig. 37, page 270.)

The chariot, having been unscrewed from the plate, is attached to the cock. It is important to make sure that the jewels, with the endstones removed, are exactly at the same distance apart as when fixed in position on the plate, the measurement being made by means of a pinion-gauge (or douzième gauge) enclosing the cock and chariot.

By employing the internal gauge described in Chapter VIII. (Art. **508**), and shown at fig. 2, plate V., this verification can be accomplished without difficulty, and the distance between the two jewelled holes can also be measured.

In its absence the interval separating the two stones must be determined either by deducting the thickness of the stones from the total external height or by some other means.

When this internal space is known, it is set off on the rule from one end, as indicated by the mark *b*. The two lines *c* and *o* are then made to indicate the space available for the axes without unduly reducing the strength of the small shell or the thickness of the collet.

Having prepared the rule it will be easy to make the corresponding cylinder, for we now know:

1. The height of the great and small slots;
2. The length of each shell, *o i* and *s c;*
3. The total length, *o c*, of the cylinder;
4. The length of each axis *a o* and *c b*.

The total length of the cylinder with its pivots will be given by merely adding the thickness of the two rubies. It is, of course, understood, that the length of an axis working between two pivot-holes is always measured from the extremities of the pivots and not from the shoulders. (If provided with the small gauge or the experimental cylinder with an adjustable axis, which are represented at C and D, figure 37, and described in articles **474** and **475**, these measurements will be materially facilitated.)

Unless a workman has had considerable experience in the making of escapements it will always be best, when the old cylinder is not obtainable, to first construct the brass rule. If he attempts to do without it he will nearly always find that he loses more time in trials, verifications, etc., than it would have taken to determine the heights accurately as above explained.

If the old cylinder is broken in two pieces, these may be

united by coating the interior with sealing-wax, and the measurements can be made when the parts are thus held in position.

To make a Cylinder.

464.—The rapidity with which any given operation can be performed depends on the method adopted in carrying it out; we have, therefore, numbered the several processes after having first arranged them in the most convenient order.

1. A piece of drawn steel, known as screw-steel, is taken as free as possible from blisters or other faults, about 5 centimetres (2 inches) in length, and a third as thick again as the required cylinder. After having straightened it, the metal is heated to a bluish-grey, and allowed to cool slowly on the bluing tray, so as to soften it. (It is well first to *prepare* the steel as explained in the article in the *Handbook* on this metal, for the work can then be performed with more facility, and the cylinder will be less distorted on hardening.) Having made a point at one end, the centre of the other is marked with a punch; this end is then drilled, care being taken to apply the oil liberally both to the drill and to the point of the lathe-head.

The ferrule fixed to the steel wire should be of an average size; the bow somewhat weak so as to remove the inside uniformly, and not to produce a boss at the bottom of the hole, or to break the drill; this should be held in rather small pliers.

The diameter of the hole should be rather less than the length of a tooth of the wheel, and when the depth is sufficient, the cylinder must be roughed out in the lathe, and cut off with a graver or cutting file.

2. The cylinder is now fixed, but not too firmly, in a thick screw-ferrule of brass in order to hold it while the broach is passed through. It will not do to use pliers for this purpose, for, besides the risk of flattening the tube, it is quite impossible to make the inside smooth.

The broach employed for enlarging the hole must be very slightly coned, and thoroughly hardened: the angles quite true and smooth, and all the faces equal. The broach is introduced alternately at opposite ends of the cylinder, and the perforation will, therefore, be slightly contracted towards the middle. When the teeth of the escape-wheel enter easily, but without play, the hole is sufficiently enlarged.

It must be remembered that when the teeth of the wheel are at all short, they must have rather more play in the cylinder, so that the shell may not be too thick; for excessive thickness must always be avoided.

3. With a view to render the tube perfectly smooth internally, it must be smoothed with a wire supplied with oilstone-dust until no scratches are observed. This operation will give some of the play necessary to the tooth, and the subsequent polishing will complete the amount. The wire having been passed through the cylinder, this is caused to rotate by rolling it against the finger or the palm of the hand, and, while continuing to roll it backwards and forwards, the wire is moved in and out in order that the stoning may take place uniformly over the entire surface, and in all directions. It must be introduced alternately from opposite ends as when using the broach.

The wire employed consists of a piece of screw-steel, about an inch of the length of which is reduced by a file until it enters easily into the cylinder. If a wire that is thin throughout its entire length, or an iron wire be employed, it will bend under the pressure applied, and the cylinder will wear into a conical form at the two ends.

A piece of steel entering the tube easily, and firmly held in the jaws of a band-saw frame, might also be employed for this process. The cylinder is moved in the manner already described, and the frame is held by its handle.*

4. When the internal smoothing is complete, the cylinder is pressed without straining on to an arbor having the least possible taper, and, rotating it by means of a horse-hair bow, the cylinder is turned down with the long point of a graver until it is perfectly round, smooth and cylindrical throughout its entire length, and just passes between the teeth without play. Continuing the rotation, it is then stoned externally with an iron file until the marks of the graver are erased. By this means some of the requisite play will be obtained.

* The methods adopted by cylinder-makers for this internal smoothing are various: some replace the saw by a suitable wire, and fix the frame in a vice; if a screw ferrule is placed on the cylinder the smoothing can be accomplished by rotating it by a bow, or by two working in opposite directions. Several cylinders may thus be smoothed at once, but it will be well to give them a slight longitudinal movement.

Others cause the cylinder to rotate along the finger, having previously affixed a ferrule, &c.; but these various methods require considerable skill, and the workman cannot do better than adopt the methods first described.

5. If the length of the cylinder is too great, it may now be reduced, care being taken to turn the extremities perfectly square; but it will be better to do this when the slot is cut in the cylinder. It is necessary to make sure, before doing this, that the shell is not too thick. If it is, the cylinder must be re-made, unless it is found possible to reduce the thickness by enlarging the hole.

6. The slot must be cut either by a square file, or on the depthing tool, employing a sharp circular cutter worked by a bow. The thickness of this cutter should be equal to the height of the slot.

The cylinder is first fixed on a perfectly true hard brass arbor introduced at the end that will form the great shell; it must be held so firmly as not to rotate under the pressure of the file. The great slot is first formed. This portion of the cylinder should descend in the narrow opening of the Jacot gauge (Chapter VIII., Article **505**) to the same figure as that indicated by the complete cylinder in the wider opening. If the measurements are made with a micrometer, the metal should be removed to a depth of very nearly five-twelfths the total diameter; the half-shell will thus measure one-half plus one-twelfth of the diameter (**340**).

The cylinder compass (Article **504**) also gives the required proportion.

But it must not be forgotten that these are the dimensions of a cylinder when finished; and, since the rounding, stoning, and polishing of the lips will of necessity remove a certain amount of metal from them, one or two degrees in excess on the gauge should be left, in order that, when finished, the cylinder may not be too open.

The great opening should not exceed one and a half times the thickness of the wheel in height, so that the oil reservoir, which is thus formed at the top of the cylinder, may keep the internal locking surface supplied with oil.

The small opening or slot should occupy three quarters of the circumference of the cylinder.

8. The faces of the shells must be filed perfectly square and flat. To secure this result various tools described in Chapter VIII. are employed.

9. To ensure that each lip is correctly formed in the manner indicated in articles **335** to **339**, *On the form of the lips*, the cylinder must be held on the extremity of a turned

brass arbor, entering only as far as the upper face of the great opening and, in using the file, great care is necessary (1) not to remove too much metal from the lips, for that would spoil the cylinder by making it too open; (2) not to slope the file in the direction of the banking slot while shaping the small lip, a very common mistake with workmen who are wanting in the requisite skill and practice; (3) not to distort the cylinder either in filing or moving it on or off the arbor; it should be taken off by pressing with the pointed tang of a file above the half-shell, and at the back.*

10. All external roughnesses are removed.

11. They are also removed from the inside by means of a wire supplied with oilstone dust, and the lips themselves are stoned until no file marks are visible. This stoning, as well as the subsequent polishing of the lips, is accomplished by means of small strips of steel and hard bronze, erroneously termed *polishing zincs*, which are of the same form and size as the files previously employed; the strips of steel are to be preferred. Should the workman experience any difficulty in maintaining them strictly flat, the part held in the hand may be bent slightly to the right or left until the most convenient form is obtained; or ruby files might be employed for this purpose.

Care must be taken not to alter the form previously given to the lips.

Hardening the Cylinder.

12. Several methods are adopted for this purpose.

One consists in suspending it by a fine iron wire with a hook at its extremity; keeping it constantly in motion, the cylinder is then held in a small flame of a candle, in the dark, in order that the tint of the metal may be more accurately observed. It is dropped into oil immediately on attaining a cherry-red colour throughout, which must never be exceeded.

Some workmen enclose the cylinder in a small brass tube, and, after heating to a cherry red, plunge the whole into oil.

One cylinder, or a number at a time, can be hardened by

* Four files, stoned on their uncut faces, are required to perform the operations here described: (1) a square extra super cut file of the same size as the great opening, so that all risk of making the engaging lip curved may be avoided, as might happen if the file were of insufficient thickness; (2) a second cut equalling file, known as a banking slot file; (3) a square smooth-cut file, bevelled at its extremity. This is used for forming the incline of the exit lip, and should have the same width as this lip, for, if larger, some difficulty will be experienced in holding it square; (4) a small, flat, smooth, and quick-cutting file for forming the corners with sharp angles.

any one of the latter methods explained for hardening the escape-wheel (**496-7**).

13. When satisfied that the cylinder is sufficiently hard it must be washed with care, since in this state it is extremely fragile; soap is employed when necessary, and afterwards it must be thoroughly dried; it is of great importance that the surface be perfectly clean and dry before tempering.

14. For tempering the cylinder a pin of brass wire is fixed in each end, being held by gentle friction without straining; the pins must not reach beyond the flat of each shell, and may project about $\frac{3}{4}$ inch externally. The half-shell is now held in a small pair of pliers, the jaws of which do not exceed the exit lip in width, and the end of the pin in the small shell is introduced into the flame until this small shell assumes a bluish-gray colour. The other pin is then similarly heated until the temperature of the great shell is raised to the same extent.

It will be found that the locking surfaces of the cylinder are by this means left quite hard. Those watchmakers who temper them to a straw-yellow, more or less marked, make a mistake; for the lips will, in that case, wear the more rapidly. As to those who temper the cylinder throughout to a blue tint, and there are very many who do so, we would only state that they are ignorant or clumsy workmen, and that they deserve to suffer more inconvenience than that actually experienced.

15. The cylinder is now held at the end of a brass wire so as to whiten the interior of the banking slot and the flat of the shells by means of oilstone-dust. It must be washed in soap and water, and dried with very great care, in order to completely remove the oilstone powder when this final smoothing is completed.

16. The inside of the cylinder is polished with coarse rouge and then the outside, after having slightly coned the great shell externally where the collet is fixed. The lips are similarly treated, first with fine rouge and subsequently with the burnisher. Finally the internal and external faces of the cylinder are finished with fine rouge.

Many of these operations can be accomplished by using small ruby or sapphire files, such as we have already referred to.

Care necessary in polishing.

465.—When polishing a cylinder that has already been cut, it must not be caused to rotate on the finger in the manner adopted in the stoning down. The internal locking surface must

first be polished by moving the polisher longitudinally backwards and forwards, with an almost imperceptible motion sideways; but it must not be allowed to leave the inside of the back of the cylinder. When this portion is nearly finished, the whole may be polished by rolling on the finger.

In order to polish the external surface the cylinder is held on an arbor placed between the points of the turns, the ferrule being held between two fingers, so as to communicate a gentle semi-circular oscillating movement to it, and along the back of the cylinder a flat polishing strip is drawn, to which a motion of rotation is given, so as to remove scratches. The polishing is then completed, using a bow in the manner already explained under smoothing.

If such precautions are not taken, the cylinder will become oval, and the corners of the lips will be too sharp, since on the open side of the cylinder the shell will offer a less surface, and it will therefore be worn more than on the back.

For a similar reason it is essential that before cutting a cylinder both the internal and external stoning down be completed.

The rouge must be very carefully prepared, and only a small quantity used at a time; the broach should be roughed with a file each time that it is dried.

Besides the necessity of avoiding all risk of distorting the lips, it is necessary that all their angles be rounded and polished; and, further, that at the point at which the tooth enters the cylinder the curve of the lip is continuous with that of the locking surface.

The cylinder must never be gripped between the jaws of the pliers, nor forced on an arbor that has not been ascertained to be perfectly smooth, for otherwise it might scratch the interior of the shell.

Other methods of smoothing and polishing.

466.—It is also possible to stone and polish the external surface of a cylinder on the depthing tool.

Two centres, adapted to the tool, are firmly connected together by a semi-circular piece of metal, so that they lie in the same straight line. By holding the metal in the hand, when the centres are in position, it is thus possible to impart to them a slight longitudinal motion to right and left.

The cylinder is mounted on a perfectly true arbor, which is

placed between the two fixed centres of the tool; between the movable centres above referred to is placed an arbor carrying a cylindrical roller. By means of the adjusting screw of the instrument, this roller, charged with oilstone-dust and oil, or polishing rouge, is brought in contact with the surface of the cylinder. Then, while one hand works the two bows in connection with the arbors of the cylinder and roller, a backward and forward movement is given to this latter by the other hand in the manner already explained.

By this means it is easy to stone and polish at the same time several cylinders mounted on one arbor without altering their figure.

It is also possible to polish the external surface of a cylinder on the lapidary's lathe. In this case the cylinder is mounted on an arbor supported between two centres which are carried by a lever capable of being gradually approached to the lap by a screw. (See also the *Watchmakers' Handbook.*)

To make the Plugs.

467.—A piece of screw-steel free from flaws, hardened in oil, and tempered to a blue tint, is used for making the plugs.

It must be turned true at its extremity, and when it almost enters the small shell its further reduction is effected by means of an iron file with oilstone-dust and oil; this is continued until it enters to about three-fourths the depth of the small shell.

The end of the plug is turned flat, and the axis roughly shaped out with the graver, its point being formed.

The face of the plug is burnished on the plug-tool (Chapter VIII., Article **506**), or, in its absence, on a screw-head tool.

Proceed in exactly the same way to form the top plug, but it must be made to enter to rather more than three-fourths of the depth of the great shell.

While turning and fitting the plugs, it is necessary to examine them from time to time with the micrometer, in order to make sure that they are fairly cylindrical. This is the form that must be given to them, for if they are at all conical, there is danger of their bursting the cylinder.

Such an effect is sometimes, though rarely, occasioned by the use of steel that is burnt, or contains flaws.

The plugs must be accurately fitted into the shells. If well formed, the least possible blow of the hammer, or merely pressure, will suffice to fix them firmly; in a contrary case,

although considerable force be used, the plug will be liable to displacement when the hollows, points, &c., are formed.

In fitting the plugs into the shells some care is required, lest, when the metal has been too much reduced, so that it enters the cylinder beyond the requisite distance, the internal locking surface should be scratched.

After the faces of the plugs have been polished, the adjustment should be such that, in an average size cylinder, the lower plug enters the small shell to a distance of about three-fourths the depth of this latter under the pressure of the hand, and the top plug to about five-sixths the depth of the great shell. This, then, shows that they will be driven under the hammer to a distance of one-fourth and one-sixth respectively.

In a short cylinder these distances will be one-third and one-fifth.

Plugging.

468.—Begin with the lower plug. The axis being held in suitable clamps in a vice, the cylinder is driven on to it by a series of gentle blows with a hammer in a manner identical with that employed for placing the balance-collet in position (**471**). The nose of the punch must be thin enough not to become fixed in the banking slot, as the cylinder would then be liable to break.

The top plug is inserted in the same way.

Some cylinder makers, having fixed the punch in the vice, rest the cylinder on it by the banking slot, and, holding the axis vertical by tweezers, strike its point gently with a hammer. Others use a verge-wheel riveting stake provided with jaws sufficiently thin to enter easily into the slot. Others again merely drive the cylinder into its position by pressing with some force on the flat of the shell with the thin end of a flat file, the plug being held in clamps in a vice as above explained. Lastly, others force the plug into its position by means of a perforated punch which freely admits the axis, the cylinder resting on the plugging punch.

To Centre the Cylinder on its Two Points.

469.—A cylinder can be centred on its points by filing them as is done in the case of a pinion. These points are then turned down with very great care in order that they be perfectly true, that is, accurately rounded co-axial cones.

Very many workmen proceed differently. Having fixed a ferrule on the balance collet, they place the cylinder in the turns, supporting the end of the shell in a conical hole of a boring plate. The point itself is thus kept free. The tool-rest must now be brought very close up, and the point turned to the requisite shape by using a horsehair bow and a wide-nosed graver, always working inwards, that is, from the point towards the cylinder itself, and never in the converse direction; for otherwise there would be some danger of catching the graver under the point. This method requires much skill and practice to avoid disturbing, and sometimes even completely displacing, the lower plug, and sometimes the cylinder is broken. It is almost the only method employed in the factories at the present day. It is essential that the conical hole be abundantly provided with oil.

470.—To diminish the risk when forming the points, the cylinder may be filled with wax. After having filled the interior with small pieces of sealing-wax or shellac, it is held lengthwise in a suitable pair of long nosed pliers, the slot being turned upwards, or a light pair of pin-tongs may be used; this is then introduced into the flame, but at a point some distance from the cylinder. As soon as the wax has melted and filled the cavity, the cylinder is immediately removed from the pliers to prevent overheating.

When the points have been formed the hollows are cut, the axes turned true of the right dimensions and coned towards the pivots. It is, however, preferable to fix the balance collet in position, and turn it true with the points before this, as by so doing some risk of breaking the axis, etc., is avoided.

The wax having been removed from the cylinder, it is now placed in the depthing tool with a view to the verification of the escapement (**429**).

The wax may be removed by placing the cylinder in spirits of wine contained in a small pan of thin red copper, and caused to boil for an instant; any wax that does not dissolve with sufficient rapidity is removed as soon as it is soft by means of a fine piece of pegwood. The spirit is again caused to boil momentarily, and, when the cylinder has been removed, it is held between the fingers, and the cleaning is completed by a brush of medium hardness dipped at intervals in the boiling spirit.

As spirits of wine is very inflammable it should be kept at some distance from the face, and if it happens to ignite blowing should not be resorted to in order to extinguish it; the vessel must be at once removed and brought against the under side of the bench, when the flame will be put out.

The Collet.

471.—The collet, formed of well-hammered brass, should be roughed out on an arbor to approximately the right height and thickness, except that where it is riveted to the balance, the height is, as a precaution, made rather excessive.

The cylinder having been fitted into the collet so that it enters about half-way, this latter is placed in a hole of the riveting stake that it enters easily but without play, and it rests evenly on one of its shoulders. The nose of the plugging punch (c, fig. 35, page 252) is now placed against the internal face of the great shell, and the cylinder gently driven into the collet by slight blows with a hammer.

The cylinder is put in the turns and the collet now finished, with the exception of the excess of metal where the balance is riveted; this must be turned down with the whole centred on the pivots or cones of the axis, so that the balance may be fixed perfectly true, flat, and at a proper height.

The pivoting of the cylinder may now be proceeded with.

To Pivot a Cylinder.

Attaching the Balance.

472.—Escapement-makers in factories are generally working on watches of definite callipers. They can enlarge the pivot-holes and allow the unpivoted axes of the mobiles to pass through them without having any difficulty in setting the escapement in adjustment. Watchmakers, however, that are occupied with the repair of watches are placed at a considerable disadvantage, for the position of each mobile is rigidly fixed by its jewels.

They can overcome this difficulty in the following manner:—

473.—Before turning the pivots it is essential to mark on the axes the extremities of these pivots, for the cylinder rotates between two plates, the endstones, and therefore these extremities are fixed.

The requisite measurement can be taken from the old cylinder, the parts being joined, if it is broken, by means of shellac.

If the broken part is not accessible, the position of each pivot, and of the balance, can be easily ascertained by using the experimental cylinder with adjustable axis, or the small gauge described in the two following articles, and represented at D and C, fig. 37.

When neither the old cylinder nor these appliances are available, recourse must be made to the small brass rule prepared when first commencing to make the cylinder (A, fig. 37).

It will be remembered that *i a* is the distance between the face of the small plug and the inside face of the chariot jewel.

The brass rule being held upright and resting on the jewel of the chariot, which must be detached from the plate, measure with a pinion gauge with thin sharp points, the distance from the lower side of the deeper slot to the outside of the jewel.

If one jaw of the gauge be now rested on the face of the small plug (B, fig. 37), the other will mark the extremity of the pivot. This point is marked with a cutting file, and afterwards carefully verified.

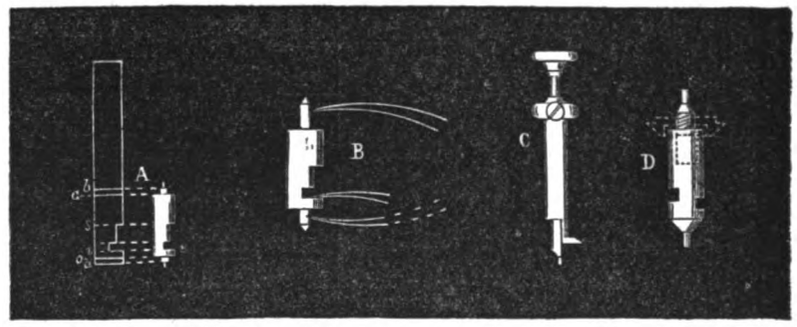

Fig. 37.

The chariot and cock are next screwed to the plate, and the distance between the external faces of the two jewels, with their endstones removed, is measured by a douzième or a pinion gauge. One point of this compass is then placed in the nick already made in the lower axis (B, fig. 37), and the other point, after being covered with rouge, will mark off the top pivot.

The height for fixing the balance can be ascertained by using a piece of brass, which is placed beneath the centre wheel, and is always separated from it by a distance equal to

the thickness of the balance plus the play necessary between it and this centre wheel.

The brass thus adapted rests on the top of the watch-plate above the lower cylinder hole, and the distance from the top surface of the plate to the outside of the chariot jewel, without an endstone, is measured with the douzième gauge.

This distance gives the interval between the extremity of the lower pivot, and the bottom of the collet shoulder on which the balance is riveted.

When the cylinder has been thus far prepared, and the height of balance determined, the pivoting, fitting of the balance, and the adjustment of the escapement will offer but little difficulty. (If any should be met with, it will be removed at once by a study of the following article.)

It is well to notice that the pivots should be turned down with the graver until their thickness is such that the final fitting can be made solely with the use of a coarse burnisher; the employment of a pivot-file, which is apt to make their form oval, is thus avoided.

When riveting the balance, the holes in the stake employed must widen downwards throughout its entire thickness. There will then be no danger of breaking the cylinder during the operation.

One arm of the balance must be set exactly over the back of the cylinder.

Tools for measuring the heights.—Practical details on Pivoting.

474.—The small instrument represented at c, fig. 37, which may be called an adjustable square, consists of a tube in which an axis, pivoted at one end, slides easily, and can be clamped in any position by a screw. At the lower end of the tube is a small pointed nose, projecting at right angles. The tool is used as follows:—

With the escape-wheel in position, the pivot of the tool is inserted in the chariot pivot-hole, and the projecting point is set exactly opposite the middle of the flat of the wheel.

The space between the end of the pivot and this point gives accurately the distance between the middle of the banking slot and the extremity of the lower cylinder pivot.

By standing the detached balance, supported on a thin sheet of brass provided with three screws to form adjustable

feet, on the plate of the watch, and setting it at the desired distance beneath the centre wheel, it is easy to measure its height above the lower pivot-hole by means of the appliance. The distance between the end of the pivot and the lower side of the nose gives the height it is necessary to set the shoulder of the balance collet above the extremity of the lower pivot.

475.—The workman may also employ experimental cylinders with adjustable axis, as represented at D, fig. 37, for this purpose, and, in spare moments, he should make two or three of different sizes.

The cylinder and lower plug are formed in one piece, in order to increase the strength; the slot shallow and in different positions occupying different heights (for the position of the banking slot is the most difficult to ascertain), and the cylinder is only perforated where the top plug is inserted. This latter can either be screwed in or slide, being fixed by a clamping screw. It may carry on its outer surface a small collet held by friction, to be used for ascertaining the proper height of the balance. But with these details the watchmaker will be able to decide for himself as to the most convenient arrangement.

476.—When the several heights have been accurately determined, and the collet turned true and smooth on its lower face, the cylinder must be covered with sealing wax or shellac, care being taken that it is thoroughly coated near the under face of the collet, while on the other hand, the face of the small shell must be left free, so that it is always possible to measure the distance between the end of the pivot and the banking slot. A small and thin ferrule, with a hole somewhat greater than the cylinder, is now held by its groove in a pair of tweezers. These are held in a flame at some distance from the ferrule, and when this is hot the cylinder is gently pressed in until the collet is against the ferrule; the two are thus firmly held together, and the rigidity of the cylinder is increased.

This arrangement enables the workman to entirely finish his work without changing the ferrule, for the pivots and collet can be completed as well as the riveting on of the balance.

The ferrule must be detached by holding it in the tweezers and heating them at a distance as before. By inverting it the the cylinder will come away of its own accord, or when a very slight pressure is applied, and it must be cleaned in the manner explained in article **470**.

THE ESCAPE WHEEL.

477.—The construction of this wheel is an operation of extreme delicacy. A watchmaker who has not had considerable experience in this class of work had better apply at the watch material dealers who keep a supply of very well made wheels of all sizes and thicknesses, or to the manufacturers, sending them by post the old wheel, carefully packed.

But for the benefit of those who are compelled to make the wheel, or of those watchmakers who, having time and patience, are anxious to improve themselves by making a complete cylinder escapement, we proceed to give the practical rules that must be followed. These rules will be of no service to practised escapement-makers, who do not always adopt the same methods or tools, and are endowed with unusual skill from daily experience; we are not writing for the very small number of those who do know, but for the very large number of those who do not, and wish to learn.

The escape-wheels used in the Geneva factories are, for the most part, roughly finished in that town; the arms are cut in the punching machine, and a very soft quality of steel, prepared with special care, is required for this purpose. They are then sent in the rough to a district of the Savoy where they can be finished at small cost.*

478.—To make a cylinder escape-wheel the workman must be provided with (1) a first-class wheel-cutting engine supplied with a special appliance for forming the U's (unless the pillar-tool is arranged for this purpose); (2) a tool for rounding the pillars; (3) a tool for forming the inclines. These tools must always be kept in perfect working order. Before using them it is well to test them on a trial wheel.

To measure the heights, thicknesses and diameters.

479.—If the old wheel is not available as a pattern, we must begin by ascertaining the dimensions of the required escape-wheel.

THE TOTAL SIZE is, according to the escapement-makers, given by the fourth wheel of the train, a sufficient play being allowed between its pinion and the heels of the escape-wheel teeth.

This mode of measuring assumes that the fourth wheel

* The Besançon district is also well known to manufacture, on a large scale, escape-wheels, cylinders, etc.

is properly proportioned; but, as we have already pointed out (**399**), in the case of very many modern callipers, we should by proceeding thus make the wheel too large.

When the total size has been ascertained, a circle, B (fig. 38, page 278), is drawn on a smooth sheet of brass of the exact diameter determined upon.

A brass disc of this diameter is now turned out on the lathe; it is introduced into the wheel-cutting machine and its rim divided into twice as many parts as the wheel is required to have teeth. This subdivision is accomplished by means of the cutter subsequently employed in forming the wheel itself. It must be specially selected, for its thickness gives the measure of that of the shell of the cylinder (**405**), that is to say, its maximum thickness is one-eighth the length of the inclined plane.

One of the brass teeth together with the spaces on either side will give, very approximately, the diameter of the cylinder. This diameter is accurately measured by means of a fine pointed pinion gauge, and a smooth arbor, passing with very slight play between its two points, can be taken to represent the cylinder. The arbor is now placed between the longer arms of the incline compass (**503**) in the correct position, that is to say, against the figure indicating the lift that it is desired the escapement shall possess (**397**). The interval between the two points of the compass indicates the distance of the circle D (fig. 38, page 278) from the circle B, the two being, as is evident, concentric.

The external circle gives the total size of the wheel; the inner one marks the portion that has to be hollowed out. It must be remembered that we are here only considering the dimensions of a finished wheel. (See the subsequent article **483**.)

In the absence of the compass above referred to, a micrometer, with jaws between which the arbor is placed, can be employed to determine the height of the incline in accordance with the explanations given in article **397**.

480.—The TOTAL THICKNESS OF THE WHEEL, when the workman has no pattern to go by, must depend upon the amount of space available without detriment to the correct proportioning of the other parts of the escapement. Introduce a small piece of brass, filed to a convenient shape, between the cock and chariot when these two are connected together apart from the plate, and by this means ascertain the exact distance

between the two jewels. Mark on the edge of the brass with the fine corner of a file; (1) the length absolutely required for each axis, in order to keep the two ends of the cylinder at a sufficient distance from the jewels and their settings; (2) the height necessary to ensure strength in the lower shell; (3) the thickness of the collet. The remaining space gives the total height of the great lip, and it will be easy to determine the most convenient total thickness that can be given to the wheel. In well planned modern watches this thickness equals about two-thirds the height of the opening.

481.—The THICKNESS OF THE FLAT OF THE WHEEL should be no more than is required for the due solidity of the arms. The *flange* (giving the width of the rubbing faces of the teeth) has the same thickness as the flat (sometimes it is rather greater when this latter is very thin), and the thickness of metal appropriated to the pillars is somewhat greater, but it will depend on the elevation of the impulse plane.

482.—With the help of these practical details any intelligent and skilful watchmaker will be enabled to make a well-proportioned wheel. But anyone who finds difficulty in accurately determining the several dimensions had better take them from a well-made wheel with the required number of teeth. The diameter, thicknesses, etc., may be increased or decreased so that the two wheels are proportional to the thickness and diameters of the watches to which they belong.

483.—*The proportions here given for the height of the teeth, the thickness of the flange, and the height of the pillars, etc., refer to a finished wheel; the workman, in roughing it out, must always remember to leave it from a third to a quarter larger, for the stoning and polishing will reduce it by at least this amount.*

It is, as we have already observed, advisable to make at least one wheel as an experiment, not merely in order to make sure of the accuracy of the proportions and to try the tools, but also to test the cutter employed in forming the U-spaces, as it is essential to know the exact depth to which they are cut before marking out the arms of the wheel.

To make the Wheel.

484.—English cast steel, known as square steel, is excellent for making escape-wheels. It is first reduced in thickness by forging, very great care being taken to prevent burning of the metal, and, when nearly of the desired thickness, is subjected

to a gentle hammering while cold, after which it is annealed. This treatment renders it very soft for working. (The article on *Steel* in the *Handbook* gives other modes of preparing the metal.)

In the factories bands of ready-prepared rolled steel are kept, and this is very easily worked with the graver.

A square piece of such metal is taken, and after drilling a hole in the centre of a convenient size for riveting the pinion, the metal is rounded with a file. This hole must be broached out with care and perfectly straight, employing for this purpose a broach that is very little tapered. The wheel is turned carefully of the required diameter and thickness on a rather short arbor, round and corresponding in form to the broach employed.

The metal is now turned down in such a manner as to leave a flange (in which the inclines are cut) projecting.

In factories the escapement-makers hollow out their wheels on a small mandril with a slide-rest, which is worked by the foot, an intermediate distributing wheel being employed. When this or an analogous process cannot be resorted to, they hollow out the interior of the wheel, which must be kept perfectly flat, and work from the centre towards the rim, using a cutter the nose of which is hooked and very carefully ground. It is necessary that only a small quantity of metal be removed at a time, for otherwise there would be some danger of distorting the wheel or bending the arbor; and if the cutter has not a good edge it will, as it were, rub the metal instead of cutting it keenly. This will render it the more apt to lose its shape in the hardening, and might even make the flat of the wheel, when it is thin, become *cockled*, and move up and down under the pressure of the tool. Or the wheel can be hollowed by the use of an ordinary graver.

The cord of the bow should pass round the ferrule in a converse direction, that is to say it must cross on the opposite side. By this means the movement of the bow will cause the ferrule to rotate backwards.

The tool-rest being so placed that the point of the graver can travel past the arbor, the interior of the wheel is hollowed out and shaped by presenting the tool to the side farthest from the workman.

A small mass of metal or boss is left untouched at the centre to ensure that the wheel may always be set true on the arbor. This is not finally removed until the wheel is quite

finished, and then the thickness must be left sufficient for riveting the pinion.

In measuring the several thicknesses a douzième gauge is employed. This will be found described in the chapter on tools (**1484**). The wheel must never be removed from the arbor by means of a hammer.

485.—After smoothing the flange with oilstone-dust, the arms are roughed out so as to admit the screws by which the wheel is fastened to the table of the cutting engine. The under-side of the wheel and the flat of the teeth are then stoned down, either on a sheet of ground glass, or a large polishing iron, or on a lap. A thick steel disc fastened to the end of an axis carrying a ferrule (fig. 34, page 246) is employed for stoning down the interior of the wheel. This disc is drilled out at its centre so as to admit the small boss in the middle of the wheel. Its face must be perfectly flat, and its diameter should be such that it just enters the hollow with a slight amount of play; rotating the axis with a rather light bow, the wheel is stoned down by holding it with the finger against the face of the disc.

486.—The wheel is now ready for cutting, and if the workman has any fear of breakage he may strengthen it with brass.

For this purpose take a brass disc rather larger than the wheel and of at least twice its thickness. A hole smaller than that of the wheel is drilled at its centre, and, with a sinking tool, this hole is increased to half its depth so as to admit the boss of the wheel without contact. The disc is now placed on an arbor and turned perfectly true on its two faces; a flange is turned on the edge, forming a shoulder, so that the wheel can be adjusted on it; and, while the interior will thus be exactly filled with the brass, the flat of the teeth will rest firmly against the projecting lip.

It will be seen that when thus protected we can mount the wheel on the table, form the U-spaces, and cut the teeth without fear of distorting it—an accident that might very easily occur without this backing if the workman has not considerable experience in the making of escape-wheels. The best makers manage to do very well without such an addition. They are thoroughly at home with their tools, and their hands are steady.

487.—The wheel is divided into twice as many parts as it is required to possess teeth, for each space represents the length of a tooth plus twice the thickness of the cylinder.

The flange must at first be only partially cut through, as indicated at *n*, *l*, *h*, fig. 38. It will thus retain sufficient firmness to avoid all danger of distortion when the U-spaces are cut, or if the arms are finished before making the inclines.

The flat circular cutter employed must be of the exact thickness of the shell of the cylinder; that is, it must be at most one-eighth the length of the inclined plane. It is necessary to slightly *dish* the two faces of the cutter.

The surface of this cutter is prepared either with a wheel-cutting engine or a chisel. It must be quick in its action, for otherwise the operation would strain the wheel.

In the absence of any better means the necessary roughness can be imparted to the edge of the cutter by pressing it firmly on a fine new file and drawing it along in the direction of the cut. Any burrs that may be formed on the sides of the cutter are afterwards carefully removed.

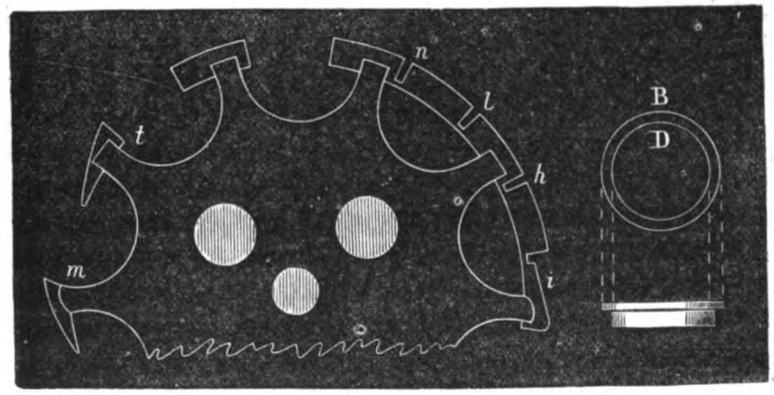

Fig. 38.

It must be placed directly in a line with the centre of the wheel and must cut sharply, a little oil being applied from time to time. The workman should remember that the heels of the teeth are to be very near the U-arms, so that, when subsequently sloped, the curves may be continuous (see *t* and *m*, fig. 38).

After partly dividing the teeth in the manner explained, and while the wheel is still on the table of the cutting engine, the inclines may be roughed out (*i* fig. 38) by means of a conical cutter or by sloping the axis of an ordinary cutter.

Pillar Tool and tool for forming the U's.

488.—This consists mainly of a sort of mandril lathe represented in fig. 3, plate IV. The arbor H carries a circular

cutter *a* (also shown at B, fig. 6), which is held in position by a clamping screw J (fig. 3). The cutter is only roughened on its outer curved surface, and any workman can make it for himself by using the sharp angle of a new file. The slide B B is caused to traverse a slot in the frame by means of the screw G. The plate *r*, seen in side elevation, is perforated to admit the cutter *a*. This plate, used in rounding the pillars, moves with the slide B B, being attached to it by two feet and a screw, the extremities of which are seen at *d*, *d'*. It can thus be detached from B when required.

The following accessories to this apparatus are necessary:

(1) The carriage shown in side elevation in fig. 4, and in end elevation in fig. 5, which, when cutting the U's, must take the place of *r*, being attached to B by the projecting piece P of the frame P *i k* that supports the carriage.

(2) A rest that is supported in the piece *f g* (fig. 3) by means of the clamping screw *h*. It is mainly of use in rotating the cutter *a*.

To form the U's.

489.—The carriage shown in figures 4 and 5 is used as follows in forming the U's: the wheel *s e* (fig. 4), after being divided into twice as many parts as it is required to possess teeth, is placed with its flat resting against the plate X of the tool where it is centred by the screw *b*, the conical point of which projects just so far as to prevent any shake of the wheel when lying flat. The plug *o* (fig. 4), also shown at A, fig. 6, is now placed within the wheel and pressed by the screw M (fig. 4): the wheel is thus rigidly fixed. The screw M, however, must allow of the wheel being rotated with some friction.

The point of the detent or guide D (fig. 5) is now placed in one of the cuts in the rim of the wheel and the carriage is fixed to the slide B B by its arm P in the manner already explained. The entire carriage rises and falls on the vertical piece *i k*, being limited in its path by the screw N. The screw G is first moved until the extremity of the cutter *a* is level with the lower side of the teeth. The exact depth to which the U's are cut can be regulated by the screw N. The position of the wheel can be adjusted by the screw F as it moves the support that carries the detent D. After making sure that all the screws are properly clamped, the workman proceeds to cut the U's, working the bow placed on the ferrule *l* (fig. 3) with his right hand, while

the left holds the carriage, gradually lowering it until it rests on the set-screw N, when the cutter will cease to act. The screw M having been eased, the guide D is raised and introduced into the next cut but one; the screws are again clamped firmly and the second U is cut, etc.

490.—At this stage the division of the wheel may be completed by introducing a fine equalling file, stoned on its faces and thinned towards the back so that it only acts with its edge, into the cuts *h*, *l*, *n*, etc. (fig. 38). The removal of a very little metal should suffice to detach the several pieces.

The heels of the teeth, *t*, are next sloped by means of a smooth curved file stoned at its edge, so that they take the curvature of the U's, as at *m*. Care is essential in this operation to avoid straining the teeth or touching their points with the back of the file, for then their lengths would no longer be equal.

Some workmen employ a special tool for this purpose which is also available for adjusting the length of the teeth when they require to be equalized. It is very similar to that employed in forming the inclines.*

Rounding of the Pillars.

491.—This operation can be performed by the aid of various tools, and the lathe described in article **488** is available for the purpose. In some cases the cutter is arranged so that it rotates round the pillar, while in others it remains stationary, and a semi-circular movement is imparted to the wheel itself. This wheel is held by the finger with its teeth upwards against a firm support (*r* fig. 3, plate IV.), which can be so placed by the adjusting screw G, that the flat end of the cutter is in a plane with the under surface of the teeth. Holding the wheel by a finger of the left hand with its flat against *r*, and working the bow which drives the ferrule *l* with the right hand, the wheel must be moved about so that the cutter rounds off the pillar in the manner required.

This cutter is driven by means of a horsehair bow, and only cuts with its cylindrical surface. A watchmaker that possesses neither a tool for making pinions nor a wheel-cutting engine,

* Escapement-makers sometimes avoid the necessity of filing the heels of the teeth by setting the cutter so that its plane of rotation passes to one side of the centre of the wheel instead of through it; the slope of the heel is thus formed in the first instance. But when such a practice is resorted to, the thickness of the cutter is no longer identical with that of the shell of the cylinder, and much skill is required to choose one of such dimensions as to produce teeth that are neither too long nor too short.

can form the cutting surface with the angle of a new smooth file. The flat end of the cutter must not be at all rough.

It is sometimes better to postpone the rounding of the pillars until after the inclines have been formed, so as to make sure that these project sufficiently from the pillars to prevent the oil from escaping.

492.—We have ourselves adopted the following form of apparatus, for with the method above explained there is some difficulty in rounding all the pillars to the same extent.

The wheel is fixed by a screw *s* (fig. 8, plate IV.), which has a large head (hollowed out below so as to admit the boss of the wheel and enable it to press on the arms), on a previously prepared brass plate P, cut away at the top in order that the cutter may have access to the pillars; the flat of the wheel, of course, is against the plate. One tooth is held by the stop *y*.

At the centre of the curve which it is desired to give to the pillars, *n* for example (which will be nearer to or farther from the pillar according as the curvature is required to be more or less pronounced), a hole is drilled. This hole fits on to a pin fixed in the plate *r* (fig. 3), occupying precisely the same position on this plate with regard to the cutter that the hole *n* (fig. 8) occupies with reference to *b*.

The action will be easily understood; P is held against *r* by a finger of one hand, the entire system P P′ being pivoted on a centre of movement *n*, and the other hand is occupied in giving motion to the cutter, which thus rounds the pillar along the arc *d d*.

After loosening the screw *s*, another tooth is brought against the tongue *y*, *s* is then screwed up, and the rounding of a second pillar proceeded with, and so on.

The diameter of the hole in the wheel is the same as that of the neck of the screw *s*. P P′ may be held by a thumbscrew or nut, and the length of its course may be limited by a set-screw.

The pillars may be stoned down in a similar manner, or by merely holding the wheel against the plate *r*, the cutter being replaced either by one that has not been hardened, but is simply roughened from time to time with a file, or by a ruby file.

In order to avoid confusion in the figures, some of the details given in fig. 5 have been omitted in fig. 4, but the one will complete the other.

Forming the Inclines.

493.—As the teeth have now been completely divided and the pillars rounded, it is a simple matter to ascertain the exact diameter of the cylinder. This dimension will enable the workman to verify the height of the teeth, employing the incline gauge, before he proceeds to form the inclines.

If the teeth are found to be too high the inclines must be cut to the requisite height without regard to this excess, and the points, which will be left large and square, must be reduced by filing inside in the manner indicated at *f* (fig. 2, plate I.).

In forming the inclines the wheel is so placed in the tool specially designed for this purpose that each tooth in turn only allows the metal to project that is required to be removed. This removal of the superfluous metal is accomplished by means of a smooth new file, which must be lightly handled. It must only be allowed to cut during a forward stroke, and should not even rub as it is brought back.

It is necessary to leave a slight excess of metal on the inclines to allow for that removed during the subsequent smoothing and finishing.

Tool for forming the Inclines.

494.—This tool is represented in figure 12, plate IV. The wheel is fixed on an arbor without a ferrule, which is then placed between the two centres L, P; the wheel thus enters between the tongues *a*, *c*, the flat of each tooth resting in turn against the tongue *a* in such a manner that no more matter projects above it than is required to be removed in forming the tooth. Having the heel of the tooth supported against the spring guide C D (the nose of which projects between the tongues to the point *a*), and holding the wheel against this stop by a finger of the left hand, the workman proceeds with his right hand to file away all the projecting matter. The file must only cut during its advance, and should be raised from the wheel when brought back.

The handle D of the guide is now pressed downwards so that its nose is removed from *a*, the wheel is rotated through the interval of a tooth and, after replacing the guide, the formation of the second incline is proceeded with.

REMARKS.—The tooth can be brought against a part of the tongues that is more or less inclined by moving the screw B, and thus altering the position of the nose of the guide.

The carriage E F, which supports the wheel, is raised or lowered by the screw J, and fixed in any position by R.

And similarly, the other sliding frame A is movable by the screw K, and can be fixed by the clamping screw M.

It is almost unnecessary to observe that when each part has been placed in position, and before proceeding to work with the tool, all the clamping screws must be firmly fastened down.

In new tools the extremity of the tongue *a* is hooked upwards to prevent the file from slipping. Some watchmakers, however, find this hook inconvenient, and remove it.

To finish the inclines and the arms of the wheel.

495.—The points of the teeth must be carefully rounded off as well as the inclines (both lengthwise and crosswise), and all sharp angles must be stoned down.

The arms of the wheel may now be finished. It is unnecessary to give elaborate details on this subject, and we would only point out that they must be thin so as to reduce the weight of the wheel, and three files in good condition are required for the purpose: (1) a rounding up file nearly flat on one side and half round on the other; (2) a crossing file, the edges of which are smooth; (3) a small flat file similar to that used for reducing the heels.

In shaping the arms the wheel may be firmly set in wax in a small collar formed with a flange on which the rim of the wheel rests; by this means it is possible to apply a greater amount of pressure, and the risk of straining the teeth is avoided.

In factories the arms are formed either by a punching machine or a special tool in which cutters are arranged to remove the superfluous metal quickly and neatly. Formerly the greater number of escapement-makers employed a kind of universal tool, by the use of which it was possible to form the U's, the arms, and the inclines, to cut the teeth and to round the pillars without displacing the wheel; but, at the present day, cylinder escape-wheels pass through several hands.

Hardening the Wheel.

496.—Various methods are adopted.

Some workmen in factories suspend the wheel by a fine iron wire doubled, the ends of which are formed into hooks, so that the wheel is maintained flat without being in any way strained. A gentle circular movement is given to it while held in a small

lamp flame, so that the heat may be uniformly distributed; and it should be in a dark room, in order that the required tint may be the more easily observed. The wheel, held perfectly flat, is immersed in oil as soon as it assumes a cherry-red tint throughout; and this temperature must not be exceeded, lest the metal be burnt. Very great skill is necessary to perform this operation with perfect success.

A Paris watchmaker used to harden his escape-wheels by heating them while completely surrounded with iron filings. He found this method to make the wheels very hard, and they were not distorted.

Some workmen enclose the wheel in a small copper box like a watch barrel without teeth and provided with a lid which enters freely and rests on the flat of the teeth. The whole is heated to a cherry red and plunged in oil.

Others place the wheel between two plates which are not screwed together but are held in position by three or four pins.

Others again employ a tube heated by a lamp. As soon as the tube is hot the wheel is held flat (with an iron wire) in the middle of it and transferred to the bath of oil immediately on assuming a cherry-red tint.

The following method is, however, best for those who have not considerable experience in hardening delicate objects.

A plate of iron, or preferably of red copper, about 5 millimetres (0·2 ins.) thick is hollowed out to a depth of 1 or 2 millimetres. It should be provided with a handle, or formed so that it can be easily held in the pliers. This plate is rested on burning fuel (glowing turf is best) in a small furnace. Immediately on acquiring a cherry-red colour it is removed, and the wheel is placed flat on its surface. The wheel will thus be heated uniformly merely from its contact with the plate (which should be lightly struck), and, when it is raised to a cherry-red tint, should be introduced horizontally into the oil.

The reason why the plate should have a certain mass is evident; if too thin it would lose its heat rapidly, and thus be incapable of raising the temperature of the wheel to the requisite extent.

A method of hardening in which the oxidation of the metal is avoided, and that may be applied with advantage to these wheels, was suggested by F. Houriet for hardening spherical balance-springs.

497.—*Houriet Process.*—"The core on which the spring is coiled is to be enclosed in an iron cylinder, and fixed in its place by a brass lid held by very slight friction. This cylinder is placed concentrically in an open iron cage mounted on supports. At the upper end of the cylinder, that is to say, at the end opposite the brass cover, is screwed a long rod terminating with a T-piece, so that the cylinder can be rotated without inconvenience from the heat. The cage thus arranged is introduced into a small charcoal furnace, taking care that as much heat as possible is applied at the circumference of the cage, so that the spring may be uniformly heated throughout; the handle is turned from time to time in order to make this uniformity the more certain. When the cylinder is sufficiently heated it is withdrawn by means of the handle, and at once introduced in a vertical direction into slightly warm water; the brass cover of the cylinder is first chilled, and as it is more expanded by heat than iron, it becomes detached and falls to the bottom of the water; the core immediately follows it, and the balance-spring is thus hardened without being in any way damaged by the air. As in this state the spring is hardened to a maximum it could not be used without at once breaking, and must be tempered with oil in the ordinary way; it then assumes a bronze tint which in no way disfigures it."

Completion and Correction of the Wheel.

498.—The inside and outside of the wheel and the flat of the teeth are now cleaned with great care, for the metal is very brittle from being hardened to its highest point; and when thus cleaned the wheel is tempered, being adjusted with slight friction on the end of a brass rod, which is held in the flame at a distance of about an inch from the wheel. This brass wire must be rotated between the fingers, so as to ensure that all parts are equally heated. As soon as the blue tint, commencing at the centre, has reached the feet of the pillars, the wheel is dipped in water, for the teeth must, so far as possible, be tempered to not more than a straw-yellow; if let down farther, although less fragile, they are not so well adapted to preserve the surfaces of the cylinder edges.

Or the wheel may be tempered by laying it on a plate previously heated, and at the same time resting on the flat of the teeth a rather heavy steel collet. As the temperature of this latter can only be raised gradually, it follows that the flat

of the wheel and the feet of the pillars may be let down to the requisite degree before the teeth themselves have changed colour. But if some teeth do not touch the mass of steel, or only do so imperfectly, these will change colour more rapidly and to a greater extent than the others.

499.—The wheel is often distorted in the process of hardening, and it may be corrected as follows:—

A disc of steel, hardened and let down to a blue colour, is turned with its faces perfectly flat and parallel, and of such a size as to fit the interior of the wheel. The wheel, fitted with this disc, is gripped in a kind of clamp, the jaws of which are provided with flat, well-fitting buttons; the whole is then held above the flame of a spirit lamp until the disc assumes a blue and the wheel a yellow tint, when it is placed in a vice and allowed to cool. Or the pressure may be applied by means of a weight, providing it is sufficient.

500.—The wheel is now placed on an arbor and delicately touched with a hard stone and oil, or in the manner adopted with duplex escape-wheels (**553**). The inside is smoothed with fine white oilstone-dust, as already explained; but a very light bow must be used lest the polisher slips, or the wheel, which it is necessary to hold with extreme care. When it is desired to impart an extra smooth surface to the inside, this may be accomplished with a wooden polisher and oil, supplemented by the residue left by the steel polisher.

The heels are prepared with a flat iron file and oilstone.

The inside of the U-spaces and the curved surface of the pillars are smoothed in the pillar tool, the steel cutter being replaced by a ruby cutter, or by one of soft steel roughened with a file. The arms of the wheel are smoothed with an iron file, and finished off with a piece of wood.

The lower face of the wheel and the flat of the teeth are stoned down, as we have already indicated, on a sheet of ground glass, a smooth piece of iron, or a lap.

501.—After verifying all the spaces, in order to make sure that none of the teeth have been distorted in the hardening, the heels are very slightly rounded with a polishing iron and coarse rouge (or a ruby file). They are polished first with medium and then fine rouge, and finally the angles, the points of the teeth, and the inclines are finished with a burnisher. These inclines, however, may, for their better

preservation, be polished and burnished *lengthwise* in the tool used for their formation. It will be remembered that the incline itself is to be what is known as *beaded*.

The inclines of the commoner class of escape-wheels produced in factories are polished in the foot-lathe with a strip of spring charged with rouge, which, being fixed to the rest of the lathe, is drawn along the inclines. It cannot be considered a satisfactory method, as it is essential that the points and heels of the teeth be afterwards polished by hand; but being very expeditious it should be mentioned.

It is fully described and illustrated by a figure in Chapter VIII. (Article **509**).

The wheel is cleaned with soap and water applied gently with a brush of moderate strength; it is then transferred first to pure water, and afterwards to spirits of wine. On removal it is delicately wiped until quite dry with a fine linen rag, followed by a soft brush.

To Pivot an Escape-wheel Pinion and set the Wheel in Position.

502.—When the old pinion is available it is easy to make a counterpart, and accurately adjust the several dimensions.

But, in the absence of such exact data, either of the following methods may be adopted for determining the position the wheel should occupy; and, with this known, it is easy to fix where to turn the rivet and the two pivots.

(1) A brass washer is placed in the escape-wheel hollow of the plate, and the wheel itself is laid upon it. The wheel is thus held against the cylinder, and it is only necessary to gradually reduce the thickness of the washer until these two mobiles are suitably placed with reference to each other.

The height of the rivet above the shoulder of the bottom pivot is equal to the distance between the jewel-hole and the upper face of the washer, a distance which may be easily ascertained by any one of the following methods.

(2) A thin disc, similar to the flat of an ordinary wheel, and cut away at *a* (H, fig. 7, plate IV.) is laid in the hollow of the plate. It rests on three small screws as feet; two of these are short, and stand in the escape-wheel hollow, while the third is longer, as it is supported in the hollow of the fourth wheel.

By turning the screws the disc is caused to lie in the horizontal plane that the escape-wheel should occupy, the projecting

piece *d* of the disc lying in the middle of the banking slot of the cylinder.

The distance between the lower jewel, in the plate, and the under surface of the disc H, is now measured with very great accuracy either by a small rule (E, fig. 7, plate IV.), in which a slot is cut, held vertically over the pivot-hole *a*, or by the adjustable square (C, fig. 37, page 270), resting its pivot in the pivot-hole. This measurement gives the exact distance between the shoulder of the rivet and that of the lower pivot.

The escape-wheel bar having been fixed in position, the workman now measures, carefully and without bending the bar, the interval between the two jewels; this gives the point at which the shoulder of the top pivot should be turned.

If these several operations have been properly performed, it only requires a little attention to ensure success.

Some watchmakers finish off the rivet and pivots before riveting the wheel in position. They use an ordinary thin screw ferrule, or one having two screws sunk in the groove and clamping between the leaves of the pinion, or a plain ferrule chamfered on either side, in which the pinion is fixed with wax.

Others first complete the rivet and the top pivot; they then fix the wheel, cement a very light ferrule, such as that shown in side elevation at F (fig. 7, plate IV.), to the flat of the teeth, and finally turn and finish the lower pivot.

The surface of this ferrule must, of course, be covered with sealing wax after it has been heated to a sufficient degree, and it should be detached from the wheel in the manner explained when speaking of the cylinder (**476**).

It is best to use a pliable black horsehair, or ordinary hair, on the bow, as the hand is less sensitive to the varying resistance of the work when one of greater stiffness is employed.

CHAPTER VIII.

MISCELLANEOUS TOOLS.—TO RE-SET A RUBY CYLINDER.

Incline Gauge.

503.—Two strips of steel, as *a b c d* (fig. 1, plate IV.), are carefully shaped with the file, and, after drawing the line E G, the position of the centre or hinge is marked at *r*; the length

r E is then adjusted to be exactly seven times *r* G (either with a compass, or, preferably, with an accurately graduated scale). The dividing lines 1, 2, 3, 4, 5, 6, 7 are marked off and afterwards the subdivisions $3\frac{1}{2}$, $4\frac{1}{2}$, etc.

The centre hole *r* is next drilled, taking every precaution to avoid displacing it, and the pieces are shaped to the form indicated in the figure, observing that each projection must coincide with the intersection of the line G E with one of the divisions, as otherwise the tool would not be correct. Finally, the two arms are united in the manner adopted in double callipers (A, fig. 1).

When the tool is closed each pair of points should be in contact.

When the cylinder is held between the points, *p*, *q*, of the longer arms, the jaws, *x*, *z*, will be separated by an interval of one-fifth (or, what amounts to the same thing, two-tenths) the diameter of the cylinder. If the cylinder is held in the points intermediate between those marked 4 and 5, the space *x z* is intermediate between one-fifth (two-tenths) and one-fourth (two-eighths) the diameter; that is, it is two-ninths.

The two pieces forming the gauge must be perfectly rigid, and the shorter arms, which can be curved or lengthened, if necessary, in order to correct any slight error, must have fine hard points.

Cylinder Compass.

504.—The gauge for measuring the half-shell of a cylinder (fig. 2, plate IV.) is formed in a similar manner, that is to say, the total length H R is divided into 56 equal parts (millimetres), and the centre of movement is placed at the twentieth such division.

If the longer arms are set so that a cylinder passes without play between *m n*, the space *a s* between the shorter ones is rather less than seven-twelfths the diameter of that cylinder (**1454**).

Had the total length been divided into 57 parts with the centre at the twenty-first, the space between *a s* would have been exactly seven-twelfths of the interval *m n*, for the proportions then would be, 21 : 36 :: 7 : 12.

The half-shell of a cylinder can also be measured with ease and accuracy by means of a dial-micrometer, if it is constructed in the manner explained in the article on *Micrometers* (**1486-8**).

Cylinder or Jacot Gauge.

505.—This consists of three strips of steel (fig. 9, plate IV.) riveted at their extremities to two cross-pieces, and separated by two longitudinal spaces *a* and *b*. The proportion between the widths of *a* and *b* is maintained the same throughout their entire length, that is to say, while the number on the side of the larger opening *b* indicates the diameter of a cylinder held within it, the same number in the smaller opening *a* gives the height of the half-shell measuring 200° of circumference.

Assume *c* to indicate the point in *b* at which the finally polished cylinder is held; if, when the great slot of this cylinder is held in *a*, it stops at the same number, the lips being finished, it is known to possess a half-shell of seven-twelfths the diameter of the cylinder.

Tools for Filing and Polishing the Shells and Plugs flat.

506.—The cylinder is held between the centres T and S (fig. 11, plate IV.), and the cylindrical piece of hardened steel R (sliding tightly on S) is raised by the screw V until nearly level with the flat of the shell. This is then made true with a flat file stoned at its edges, and the filing is continued until the file no longer *bites*. The polishing is accomplished in an analogous manner.

When using the apparatus shown in fig. 10, the cylinder is supported on the small fixed punch *d*, the block A, which slides on B, is then adjusted by means of the screw G until the flat of the shell just projects above the surface of the hardened block A.

This appliance should be provided with a second axis, similar to B, except that it is terminated by a small screw vice in which the plug can be fixed by its axis. The block is adjusted so that the face of the plug just projects above it, and this is then filed and polished.

Grammaire or Divided Plate.

507.—This consists of a flat brass disc, divided into 360 degrees, and bearing the marks indicated in fig. 1, plate V.

Let it be required to mark the lifting points of an escapement: the balance is laid on the side A, and the circular arcs thereon render it easy to place it central. With a pinion gauge the number of degrees in the lift (say 40° or the arc *a c*) is measured, and this distance is marked on the plate of the watch accurately below the balance, which must be placed in position.

The Grammaire may also be used for tracing out any angles on a body fixed on its surface, providing it does not exceed the plate in extent.

The reverse side B, having at its centre a screw with a conical head, is used for accurately crossing out the arms of wheels, balances, etc. After fixing the wheel at the centre by means of this conical screw, the several lines are drawn with the rule indicated by dotted lines, which is placed against the divisions of the circles 5, 4, 3, etc., according as it is desired that the wheel shall have either of these numbers of arms.

It is almost unnecessary to observe that when it is required to form an odd number of arms, the plate must be subdivided into twice that number of equal parts; this fact is indicated on B.

Internal Gauge.

508.—This tool, which is used mainly for measuring the interval between a pair of jewels, or the relative diameters of a barrel arbor and the interior of the barrel itself, is well known to watchmakers.

The instrument we have employed for determining the distance between the cock and chariot jewels is constructed as follows:

A strip of steel *b i d* (fig. 2, plate V.) carries at its end *b* a small screw *a*, and at its opposite extremity is fixed a graduated arc of a circle *d f*. The finger *h* is centred at *i* and its point is turned upwards as shown at *c*.

In using the instrument the portion *a d* is placed on the plate so that the screw *a* is above the chariot jewel, and it is rotated until just in contact with the stone. The cock is now screwed in position and the two points of the instrument are introduced between the jewels so that, while *a* touches the lower one, the point of *c* rests against the flat face of the cock jewel. The weight of the finger *h* will suffice to maintain contact, and, after the division to which it points on the scale has been noted, the gauge can be removed and the finger fixed in the same position by the screw *h*.

Incline Polishing Lathe.

509.—This is represented in figs. 3 and 4, plate V., and is no more than an ordinary lathe specially arranged for the purpose.

The mandril, M (fig. 3), carries a small chuck (indicated by the dotted line *n n'*) or a tapered arbor *r s*, on which is fixed the

escape-wheel whose inclines require to be polished. The slide *a b c d*, shown in side elevation in fig. 4, carries an arm, R, as usual.

This latter piece can be caused to slide in its support by a screw *f g* (fig. 4). The vertical rod H, which replaces the ordinary T-rest, is perforated at *i* to receive the axis of the irregular shaped piece *j k l*, the portion *x* of which projects at right angles to *j k*. An arm *n* carries a screw forming an adjustable stop to limit the motion of the arm in a vertical plane as required, and it can be fixed by the clamping screw *p*.

A block *m* is pivoted in the projection *x*, being held by slight friction, and it can be rotated by the pin *t*. A piece of watch mainspring, *o p m*, about two inches in length is fixed to *m* by a screw.

The instrument is used as follows:

The appliance represented in fig. 4 having been fixed on the bar of the lathe, and the escape-wheel placed firmly on the arbor *r s*, the slide is adjusted so that the spring *o m* is just above the wheel. When the several screws have been clamped a sufficient quantity of polishing rouge is placed on the under face of the spring.

The wheel is caused to rotate by a treadle and intermediate wheel, and at the same time, while gently pressing the spring against the teeth, an oscillatory movement is given to the piece *m* by means of the pin *t*. Such a motion is essential, for without it only the middle of the inclines would be polished since they possess a beaded form.

Remarks.

Workmen differ as to the exact arrangement of their tools, but the form adopted is always a modification of that above described.

Some practice is necessary in order to ascertain the most convenient rates of motion as well as the best length and thickness of spring.

The inclines can be polished on the ordinary finishing turns. In this case only the portions *j k l n*, and *o m m′* are required; but *j k* must be prolonged to *u v*, so that the hole which replaces *i* corresponds with the middle of the wheel, and the arm can thus be supported by the right-hand centre. The arm *n* is made of such a length that its screw can rest against the bar of the turns.

The left-hand centre carries an *idle* pulley, driven by an intermediate wheel, which communicates motion to the arbor carrying the escape-wheel by means of a small screw carrier; this arbor is supported between the centres of the turns and all the conditions indicated in fig. 4 are thus complied with.

When the wheel is riveted to its pinion, this latter is the only method that can be adopted. The lathe is provided with two small chucks similar to that indicated by the dotted line *n n'* (fig. 3) and perforated so as to admit the axis. The wheel having been centred by means of a pump cylinder is fixed to the chuck by screws.

To make the Setting of a Ruby Cylinder.

510.—A ruby cylinder is perfect from the point of view of wear, but its construction is difficult and costly, involving the labour of a skilful lapidary as well as of a highly intelligent watchmaker, and, even when made in exact accord with the requirements of theory, the rate obtainable with such an escapement is in no way better than that of a steel cylinder: we, therefore, in the first edition of this work omitted to give any details relating to it. We were further influenced by the fact that it is nearly always advantageous to replace a broken ruby cylinder by a well-made steel one, taking care, however, to make the internal drop slight and the shell thin (**404**).

Nevertheless, in view of certain special cases, and considering that the details of its construction may be of service in various branches of horology, we will proceed to give them. The method adopted by Perrelet, for an account of which we are indebted to M. Redier, will be followed.

511.—The steel framework of a ruby cylinder is known as the *setting*, and the ruby or cylindrical shell which surrounds and is cemented to the setting, is called the *body*.

The setting is constructed as follows:

A piece of square steel 4 mm. (0·157 ins.) in the side, and 5 or 6 mm. (0·2 in.) long, is perforated with a smooth hole having the exact diameter of the inside of the ruby cylinder.

It is turned down to the form indicated at A' (fig. 5, plate V.), the top remaining square as at first.

The diameter of the tube, after it has been smoothed, must be exactly that of the required ruby cylinder.

The arbor on which it is turned must be introduced at the lower end of the tube A'.

The extremity is filed to a blunt point as indicated at A; it is then hardened and the edge sharpened, thus constituting a sort of trepan.

A second trepan is made similar to the first, but sensibly smaller both in internal and external diameter; its several dimensions need not be so accurately adjusted, as will be subsequently gathered. They should, however, be approximately as follows: externally it should pass easily into a hole that admits the cylindrical portion of the tool A; internally it should differ from A by the thickness of the internal edge of the groove that supports the body; in other words, it is such that the lower plug fits it.

The use of these trepans will be presently explained.

When they are ready for use, a piece of soft round steel, of sufficient thickness, is perforated with a hole having the same diameter as that in the second of these tools, and it is then fitted on an arbor.

This tube is turned down at *i* (fig. 5), until it fits the trepan A accurately. The workman is now provided with three pieces, A, B, and C; the trepan, the future setting, and the arbor.

Having placed the trepan on *i*, and adjusted the arbor between the centres of the finishing turns, the trepan is held in the fingers and pressed against B while the arbor carrying it is caused to rotate, thus forming a cylindrical groove as indicated in fig. 6. This groove must be of such a depth as to admit the body and the top plug, an allowance being also made for the slight reduction in its length that must occur in the subsequent processes.

Before removing A, it should be rotated in the groove so as to make sure that the bottom is as smooth and true as possible; the portion of metal that guided the trepan and projects beyond the line *f g* (fig. 6) is then removed with the graver.

The second trepan is now placed on the end of the arbor, which it fits easily but without play, and, employing it exactly as in the first case, the thin internal shell is removed, with the exception of a projecting edge at the base, so that the metal is left as indicated by the portion of the drawing above the line *a c* (fig. 7).

In this groove the body will be fitted, and above it the large plug which also forms the balance collet. The lower plug is fitted in the smaller perforation (at *s*).

If not already done, as would be more prudent, the lower portion of the setting must be turned down externally, as shown at *s c a* (fig. 7), and the entire surface must be polished with fine rouge, especially the shoulder.

The openings must now be formed, employing for this purpose special fine cut thin files. Commence by making the great opening, which appreciably exceeds a half-circumference (fig. 8).

The smaller opening corresponding to the body is next cut and finally the banking slot; the setting is now as shown in fig. 9.

Lastly, in order to ensure a sufficient exposure of the entrance and exit lips of the body, the corners (*o*, *h*, fig. 9) are bevelled, and the setting thus assumes the form *b h d o* (fig. 10) when it is ready for hardening, providing the several operations have been accomplished without in any way straining it.

The metal is hardened by any of the processes described that are applicable to delicate workmanship.

It can be tempered to a blue colour, or even slightly beyond that point since no part is directly acted upon.

When the setting has been strained it must be restored to the proper form by the process described at the end of this article.

The polishing of the setting is one of the most delicate operations of horology, if indeed it is not the most delicate. We will proceed to indicate the precautions that must be observed in this process as well as in that of inserting the plugs.

After the internal part has been made as clean as possible, and the tube that receives the small plug has been stoned down, an arbor is passed through the setting.

All the several openings that have been formed are filled in with sealing wax or shellac, and the surface is carefully smoothed down, thus restoring it to the form *t s* (fig. 7).

At the upper end of the setting a brass disc is adjusted perfectly true with the axis and level with the face of the shoulder; it will serve as a guide for the polishing file while working on the lower portion of the setting and on that part of the top which is not cut away. The polishing in this case, and indeed in all others, must be quickly completed in order not to alter the form.

Lastly, the upper circumference is polished and the wax

removed. All the surfaces should now be highly polished. Practice alone will ensure perfect success.

The lower plug T′ (fig. 10) is of the usual form.

The upper plug T has a double shoulder and forms the collet. The slight depression at its face, in conjunction with the side of the great shell into which it is driven, forms a groove to receive the upper end of the body.

The portion of T that is driven into the setting is, of course, rather larger than the external diameter of the ruby body or half-shell of the cylinder. The actual face of the plug corresponds in diameter with the inner surface of this half-shell.

It must be clearly understood that the part *o* (fig. 10) is to be so far reduced in height that it passes freely between the flat of the escape-wheel and the under surface of the teeth; the ruby cylinder thus renders necessary a wheel with higher pillars than is employed with the ordinary one.

The plug is held in the shell by simple friction, but the body must be cemented in position. The plug is driven in while the cement is hot, and it is still possible slightly to displace the ruby body (*i c s* fig. 10), if found to be incorrectly placed.

512.—When a setting is strained, it is usual to re-dress it as follows.

When no balance is fixed on the cylinder a large thin disc is fitted truly on the great shell, practically constituting a balance.

The small shell is now firmly fixed in a hole in a horizontal plate to which it must be accurately perpendicular, and a strip of red copper, filed at its extremity like a screwdriver, is introduced into the opening whose pillar is strained. This copper rod is now heated, and as soon as the pillar commences to change colour it is bent, employing the copper as a lever. The effect is made evident by the motion of the disc, and this should be continued beyond the position in which the disc is parallel to the plate; for it is essential to somewhat overdo the correction because the pillar will contract slightly when not subjected to the pressure of the lever. This is an operation of very great delicacy.

Some watchmakers are extremely skilful in re-dressing cylinders in this manner, even if merely supporting the balance on a plate or between the fingers, but many who attempt the method only succeed in breaking the cylinders.

DUPLEX ESCAPEMENT.

CHAPTER I.

Preliminary.

513.—The first crude suggestion of the duplex escapement seems to have been made by Dutertre, a clever French watchmaker; but Pierre Le Roy was the real originator of this mechanism, having first constructed it about the middle of last century. He soon, however, abandoned it in favour of the detent escapement, and this fact has led some to infer that he had not formed a favourable opinion of the duplex, whereas it only goes to show that Pierre Le Roy, with that profound genius and extraordinary discernment which placed him so far in advance of his contemporaries, had foreseen that the detent was the only form of escapement that he could employ for accurate timekeeping.

We shall not stop to discuss the assertion of some authors to the effect that the escapement now under discussion was the invention of an English watchmaker named Dupleix, very little known even among his own countrymen, and who, moreover, if he made the discovery, did so long after the date of the French invention: one simple fact which appears hardly open to argument is that the Latin word *duplex*, signifying *double*, was applied to this escapement because, when first introduced, there were two escape-wheels, producing a double effect.

It is a singular fact, and worthy of mention, that this French invention was much admired in England, whereas the cylinder escapement, of English origin, was generally preferred in France.

In our first edition we attributed this circumstance to two causes: the caprice of fashion, so much thought of in France, that favoured thin watches, to which the cylinder escapement is the better suited, and the practical character of the English, who prefer to look upon a watch as a timekeeper rather than as a mere bit of jewellery. Further inquiry, however, has convinced us that we were in error, for a very great number of excellent thin watches have been sold in England. If in that country the public became disgusted with thin watches sooner than with us, it was only due to the fact that the English workmen, accustomed as they are to watches of larger dimensions, were, as a whole, unsuccessful in their management of the smaller class.

In England, thanks to the frequent use that has been made of the duplex escapement, many workmen are to be found who thoroughly understand its principles. But in France, watchmakers who are thus specially skilled are rare; indeed, only a few know its principle and mode of action, and they are even divided upon these points; if we further mention that the duplex has too often been employed in watches of insufficient thickness, it will be very evident why it is less thought of here than abroad.

For uniformity it is superior to the cylinder escapement, and we have numerous examples of duplex watches which, after going for a very long time, give the time within one or two minutes or even a few seconds a month. But the escapement must be of first-rate workmanship throughout. Its principal faults are a liability to setting, the delicacy of the balance-staff, and a rest very far diverging from the tangential position. At the same time it must be observed that the risk of setting can be confined within very narrow limits; that a staff broken otherwise than by a very severe blow is seldom met with; and, finally, experience has proved that the wear of the pivots, the most common result of an untangential rest, can be prevented by making them hard and highly polished, the pivot-holes being of real rubies finished with great care and always well supplied with oil. The motive force employed is greater than that required with a cylinder escapement, and while it must be sufficient it must never be excessive. This is often found to be the case in English watches, a fact which explains their wearing more rapidly than French watches of similar make. On the other hand, the motive force of these latter is often deficient in consequence of the watches being too thin. Without a good average motive force (intermediate between that adopted in England and France some years ago) it is hopeless to expect that this escapement will give entire satisfaction.

One author, who classes the duplex as inferior to the cylinder escapement, says, in referring to the former, that, "considering what results one anticipates from the duplex, it may be regarded as a deception." This statement is evidently either an error or an exaggeration, which shows that the author, without having experimented enough on the subject, sets down errors resulting from bad workmanship as being inseparably associated with the principle or mechanical

arrangement of the escapement. All those who have practically studied the duplex assert that when made by a competent watchmaker it can be relied upon as a timekeeper for a long period, and its rate is better than that of a cylinder escapement; it is well known that, in small carriage clocks the duplex escapement has hitherto given the greatest regularity.

If the reader assumes from the foregoing paragraph that, in defending this form of escapement against the unjustifiable discredit that has been cast upon it, we are expressing a wish that it should be more generally adopted, he will fall into an error; for we unhesitatingly confess to preferring the lever escapement which, while possessing an equally good rate, is not liable to set, is more substantial, and involves less absolute accuracy in both repair and construction.

Denomination of the Several Parts.

514.—The principal and distinctive parts of this escapement are:

1. A small cylinder of ruby termed the *roller* (*a*, figs. 1 and 2, and *b*, fig. 3, plate II.), carried by the balance-staff, to which it is cemented by shellac, being further held in position by a small collet, termed the *plug* (*b*, fig. 2), that is driven on to the staff. A notch or slit (*a*, fig. 1 and 4*d*, fig. 3) is made lengthwise in this roller of such a depth that the points of the teeth cannot reach the bottom.

2. A lever arm is also carried by the balance-staff (R′ figs. 1 and 2, plate II.), called the *hook*, *impulse pallet*, or simply *pallet*, which receives the impulse required to maintain the motion of the balance.

3. A wheel provided with a double row of teeth: one set are long and tapered, lying in the plane of the wheel (A, B, C, fig. 1; E, *d*, L, fig. 3); the others are short triangular prisms and resemble pins projecting from the flat of this wheel (*s*, *p*, *o*, figs. 1 and 2; R and V, fig. 3, plate II.).

Figure 2 of this plate shows the balance-staff with the collet, impulse pallet R′, the roller *a* and the plug *b* in position.

Formerly the escape-wheel pinion carried two superposed wheels, which, deriving their names from the difference of diameter and their distinctive uses, were known as the great or *locking* or *resting wheel* and the small or *impulse wheel*. These terms have been retained in the single wheel as made at the present day.

Action of the Escapement.

515.—Consider the front of the balance as turning towards the left of the observer, that is let the balance be turning from z''' towards D (fig. 1, plate II.): a locking will occur since the tooth D is supported against the roller at k, and this locking will be broken by a slight jump or recoil when the point of the tooth drops into the slit a. This is all that occurs during the vibration which is said to be *dumb*.

When the balance, brought back by the balance-spring, is turning towards the right, the tooth A (in the same figure) descends into the slit a of the roller, and, as it presses against the right side of this notch, pushes it forward (tooth B) during a short period of the vibration, finally escaping from it (tooth C) at the instant at which the pallet z is in the position z'. Since the wheel is now free to move, the impulse tooth n falls against this finger z'', and, pressing it forward with considerable energy, accomplishes a *lift* which continues until a locking is again caused to occur through the succeeding tooth falling against the roller (as indicated by D k).

The balance is soon brought back by the balance-spring and another dumb vibration occurs (while turning to the left); so that the two mobiles are brought again into the positions indicated for the tooth A, and such a series of movements occurs during each vibration.

The pallet does not touch the impulse teeth near it while the locking is taking place. Contact can only exist when the lift occasioned by the locking tooth has brought the impulse tooth within a certain distance of the line of centres.

An impulse, it will be seen, is given at each alternate vibration; the tooth does not escape from the roller until a backward and forward movement of the balance have occurred, and the lift therefore only occurs on one side.

This entire lift consists of two distinct portions: (1) the *small lift* or the circular arc traversed by the balance while the locking tooth is in the roller notch; (2) the *great lift* which commences immediately on the termination of the small lift and is measured by the circular arc passed over by the balance while the impulse tooth remains in contact with the pallet or hook.

These two successive lifts are only separated by a very small drop which is essential in order to ensure certainty of action; this is known as the *first drop* to distinguish it from the second drop after the great lift has taken place.

Proportions in Vogue at Different Epochs.

516.—TAVAN.—"As the diameter of the ring which surrounds the balance-staff (the roller) is reduced, the resistance which it opposes to the motion of the balance in friction during the rest will become less; the recoil will also diminish as the dimensions of this cylinder and of the notch are reduced. (An examination of rollers constructed in Geneva in Tavan's day shows that he regarded as a convenient diameter not more than about a fifth of the distance between the points of two teeth.)

"As the depth to which the tooth pitches in the notch is increased, its action will be rendered the more certain. The best results seem to be secured when this notch is one-sixth of the radius (that is, one-twelfth the diameter) of the roller; and its width must be no more than is required to ensure the passage of the tooth in order that the recoil may be reduced.

"The small lift commences about 55° before the impulse tooth engages with the pallet."

Tavan further observes, when speaking of the great lift, that it may be as much as 45° or 50° without inconvenience. We shall subsequently see that the exact amount of this lift depends solely on the proportions existing between certain parts of the escapement. It may be noticed that, according to this author, the two lifts together extend over an arc of from 90° to 100°.

517.—JURGENSEN.—"The diameter of the roller should be a third of the distance between two teeth of the great or resting wheel.

"The lifting action of this wheel on the roller (that is, the small lift) extends over an arc of 20°."

It is advisable that a drop of 10° occur between the escape of the great wheel from the roller and the engaging of the impulse wheel with the pallet.

The lift occasioned by this latter should measure 30°. Thus the impulse pallet should have such a length as to produce a total lift of 40°, of which 10° will be expended as drop; during the remaining 30°, therefore, it will be acted on by the wheel."

To satisfy this condition it is necessary that the diameter of the path of the impulse pallet be:

For a	12-tooth wheel,	$\frac{3}{4}$ or $\frac{9}{12}$	the diameter of	the escape-wheel.
"	13	"	$\frac{9}{13}$	"	"
"	14	"	$\frac{9}{14}$	"	"
"	15	"	$\frac{3}{5}$ or $\frac{9}{15}$	"	"

518.—MOINET (Data given by M. Ruggien).—With a 15-tooth wheel,

"The great wheel must not pitch in the roller notch more than about one-twelfth the diameter of this roller.

"The great lift extends over an arc of 30°.

"The two diameters are to each other as 3 is to 2; so that, if the great wheel is 6 lines in diameter, the small wheel must be 4 lines.

"The diameter of the roller is one-sixteenth of that of the great wheel (a very little more than two-sevenths of the distance between the points of two teeth).

"The roller notch must occupy an arc of 30° in addition to the rounding of its lips, which is about 10° for the two.

"The ratio of the arm that communicates the great lift (the pallet) to the radius of the impulse wheel may be as 3 to 5, that is to say, if the distance between the centres of the wheel and balance be divided by 8, five of these parts must be taken as the radius of the impulse wheel and three for the length of the pallet." (The pallet would then pass without receiving any impulse. If this is not a clerical error we must conclude either that M. Ruggien has expressed himself badly, or that Moinet did not notice the fact.)

"The balance should traverse an arc of 20° at least, and not more than 30°, between the passage of a tooth of the locking wheel into and out of the roller notch. The drop of the impulse tooth is sometimes as much as 10°, but there are authorities who advocate only 4° or 5°.

"When the balance-spring is in its neutral position, the great tooth should be in the middle of the roller notch, which must thus traverse the same arc to right or left in order to allow this tooth to escape."

We will conclude these extracts with the following:

"Some kinds of friction at the centres of movement appear not to appreciably interfere with the freedom of action, notwithstanding that experience would lead one to anticipate the contrary." After giving a case bearing on this assertion that had occurred in practice, Moinet adds: "What appears to be logically correct does not always occur in nature and experience is the more certain guide."

519.—M. WAGNER.—"With a 12-tooth wheel I have preferred the following proportions:—Ratio between the

diameters of the two wheels, 3 to 2;—great lift, 30°;—small lift, 50°;—diameter of roller, two-sevenths of the interval between the points of two successive teeth;—drop of the impulse teeth on to the pallet, 6°.

"The length of the impulse pallet follows from the extent of the great lift, the ratio between the diameters of the two wheels and the number of teeth. When these several dimensions are as given above, the length of the pallet is about five-eighths the radius of the impulse wheel. This length and the amount of the great lift vary with the size of the impulse wheel and the number of its teeth: to determine it accurately the escapement should be drawn on a large scale.

"The neutral position of the balance-spring should coincide with a point between the great and small lift in order to prevent setting.

"This escapement admits of a supplementary arc of no less than 260° on either side, which, together with the two lifting arcs, gives a total arc of at least 600°."

520.—GANNERY.—The impulse wheel occasions a lift of 45° measured as usual at the axis of the balance, during which period the wheel itself moves through the interval between two teeth less 6°; this allowance is essential in order to ensure the proper action of the escapement.

A motion through 15° should suffice to liberate the balance. In carriage clocks or such as are liable to violent shocks nearly 25° are necessary.

The diameter of the great or locking wheel is to that of the roller as 1000 is to 59.

The diameter of this latter is then about two-sevenths of the distance between the points of two successive teeth.

521.—M. WINNERL.—Lift, 60°. The real lift is, however, not more than 45°, the remainder being taken up with drops and the motion of the locking wheel while held in the roller notch.

The balance ought to traverse an arc of 15° from the neutral point of the balance-spring in order that the locking tooth may escape.

The diameter of the roller should be reduced as much as possible, but without going to an extreme, for otherwise the beneficial effect of the pressure which the locking wheel exerts on the roller would be lost, and, further, the entire mechanism

would become too delicate. In watches of ordinary size the diameter of the roller should be about a sixth of the distance between two locking teeth; but as we increase the size of the escapement this proportion may be reduced. On the other hand it must be noticed that in small watches it is often found impossible to reduce it below a quarter of this interval.

"As a rule the roller notch is made too narrow; that is to say a tooth can only pass with very little play; this is a mistake because the teeth must be pitched very shallow in order to escape from the notch by a movement of 15° from the neutral point, and when they are in the least worn they will enter very easily."

PROPORTIONS AND MISCELLANEOUS DATA.

522.—These have been gathered from the works of authors other than those above quoted or from the best makers of this escapement on an extensive scale.

Ratio between the two wheels; 3 to 2 and 4 to 3.

Ratio between the radius of the impulse wheel and the length of the impulse pallet; 4 to 3, 5 to 3, etc.

Size of roller as compared with diameter of locking wheel, one-sixteenth, one-twelfth, etc. (The twelfth is about equal to two-fifths of the distance between two teeth, the sixteenth is very nearly two-sevenths of this distance.)

Size of the roller as compared with the distance between two teeth, a quarter of their interval apart.

Opening of the roller notch, 30° plus 10° as allowance for the rounding of the edges.

The neutral point of the balance-spring corresponds to the middle of the small lift.

Lastly, P. Dubois states in one of his works that, according to some watchmakers, the diameter of the roller should be one-twelfth of the interval between the points of two locking teeth. This is doubtless a printer's error, for if of such dimensions the roller would be no more than an ordinary pivot, which is evidently out of the question.

Table of the Proportions recommended by different Authors.

523.—It will be well to arrange these various dimensions side by side to facilitate comparison.

Ratio of Diameter of Roller to distance between points of two teeth.

Tavan	about	$\frac{1}{6}$	Maximum, $\frac{2}{5}$—minimum, $\frac{1}{6}$.
Jurgensen	$\frac{2}{6}$ or	$\frac{1}{3}$	
Moinet	rather over	$\frac{2}{7}$	
M. Wagner		$\frac{2}{7}$	
Gannery		$\frac{2}{7}$	
M. Winnerl	between $\frac{1}{6}$ and	$\frac{1}{4}$	
Experienced Men	Practical $\frac{2}{6}$ or	$\frac{1}{4}$	
		$\frac{2}{5}$	

Length of Impulse Pallet.

Jurgensen.—The quotient obtained by dividing 9 by the number of teeth of the wheel; taking the radius of this escape-wheel as unity.

Moinet.—With a 15-tooth wheel take $\frac{3}{8}$ of the distance between the centres of movement.

M. Wagner.—With a 12-tooth wheel take $\frac{5}{8}$ of the radius of the impulse wheel.

Practice of the Factories.—$\frac{3}{4}$ or $\frac{3}{5}$ of the radius of the impulse wheel.

Ratio of diameters of the two wheels.

According to Jurgensen, Moinet, and M. Wagner—
as 3 is to 2.
„ „ good practical escapement-makers as 4 is to 3.
[or as 7 is to 5.

Pitch of Tooth in the Roller Notch.

It should be such as to produce a lift of about

50°	according	to	Tavan.
20°	„	„	Jurgensen.
20° to 30°	„	„	M. Moinet.
50°	„	„	M. Wagner.
30°	„	„	M. Winnerl.

Position of the Impulse Tooth.

Jurgensen and Moinet recommend that it be midway between two of the large teeth. Others place it somewhat to the right; others again, a little to the left.

Observations on the above Table.

524.—It will be seen that authors and practical men are no more agreed as to the exact proportions that should be observed in this escapement than they were in the case of the verge and cylinder escapements. Thus with the diameter of the roller, as with the height of the incline of the cylinder escape-wheel tooth, some recommend just double the amount advocated by others, and the remaining dimensions are

analogous. How can the practical watchmaker ever be expected to have clear and precise ideas on the subject of the duplex escapement when he finds such disagreement among authorities who have made this particular subject a speciality? And yet many of these contradictions are only apparent.

They are due to the fact that, not having any fixed theoretical basis to stand upon, each author has put forward his own particular arrangement, when it has proved satisfactory, as a *general* and even invariable rule, whereas it is nothing more than one individual case, a mere empirical datum.

CHAPTER II.

PRINCIPLE OF THE DUPLEX ESCAPEMENT.

525.—With the duplex as with the cylinder escapement, uniformity of rate can only be secured by a judicious application of the principles laid down in the theory of escapements with frictional rest.

In both the balance, that is a movable mass which alternately drives and is driven, is influenced by two forces: one accelerates its motion and the other retards it. To this double effect must further be added the action of the balance-spring.

The ratio between the several forces that are thus combined should be such as to render the escapement as far as possible insensible to the thickening of oil and variations in the motive force; at the same time securing a sufficient freedom of action to the balance-spring when these variations occur.

With regard to the discussions which so often take place at the present day on the subject of large and small wheels, of thick or thin rollers, they are valueless, being founded on no sound theoretical basis. There is no such thing as either a large or a small wheel, a thick or a thin roller; there is only one size of balance and roller that is well suited to good working, and this can be experimentally determined.

The point of supreme importance in a duplex escapement is the angle of lift. When this angle has been determined it is easy to ascertain the remaining elements, either graphically (**548**) or by calculation, or by experiments such as those already indicated in discussing the cylinder escapement.

These points will be made perfectly clear by the following

articles providing the reader has clearly understood the theoretical considerations which precede; if he has not done so, however, he will experience some difficulty.

Size of the Roller.

526.—If two duplex watches be examined, which are identical in every particular except that the roller of one is very small while that of the other is somewhat large, it is at once evident that in the first the balance is more free and the arc of vibration is greater. This must clearly be the case, for the friction is of less amount, the lever by which the opposing force is applied is shorter, and the recoil is diminished in proportion to the reduction in width of the roller notch.

This difference in the mode of action of the two escapements confirmed the opinion that the roller must be as small as possible, at the period at which long arcs of vibration were regarded as the universal panacea; no notice whatever was taken of the irregularities that time introduces in the rate (a subject indeed which is always much neglected); and such a proportion, that after all is no proportion for it fixes nothing definite, has for a long while misled watchmakers.

527.—As the size of roller is one of several elements in the mechanical combination we are discussing, it must be deduced from the actual state of the machine taken as a whole; that is from the absolute push which has to be neutralized by the pressure on the locking surface. The effect of this pressure will, of course, increase and decrease in proportion to the radius of the roller.

In thin watches the impulse is of slight amount; the radius of friction must then be reduced, and it is nearly always essential to have a small roller.

The converse is the case with thick watches.

This explains the unsatisfactory character of many duplex watches of Swiss manufacture.

The watchmakers of that country were fond of employing this escapement in thin watches, and in such the motive force falls off as the oil thickens. With a view to neutralize this inconvenience they reduced the roller to a minimum, and the circumstances justified them in doing so. The fashion for thin watches went by, and they were made of much greater thickness; but the manufacturers made the grave error of retaining in these watches the proportions that had stood them in good stead in those formerly required. Two results followed: (1) a

deficiency in the correction produced during the rest; (2) arcs of too great extent, for they often exceeded 600° (**566**).

Practical objections to a small Roller.

528.—(1) It is more fragile, and must be constructed with extreme care.

(2) Unless the balance is heavy the escapement is more apt to allow two teeth to pass at the same time if a shake occurs and especially if the wearer rides on horseback. As is well known, a heavy balance increases the risk of breakage of the staff and pivots.

(3) The axis passing through the roller becomes extremely thin and this renders the staff still more fragile when it is already too much so. And further, since the duplex must be rather thick and possess a considerable motive force, the staff is the more liable to bend in consequence of its thinness; if this does not actually break the roller, the disturbance of the axis with a kind of vibratory movement interferes with the timing and is sure to loosen pieces of cement.

(4) The slightest want of truth in the teeth of the wheels will affect the action; and the difficulty of obtaining a duplex wheel that is divided with absolute accuracy is well known.

(5) The notch is cut to a very slight depth for fear of cutting through the roller; the most minute particle of extraneous matter in this notch, even a little oil that has thickened, is enough to impede the points of the teeth and so alter the rate.

(6) The corners of the notch must be almost sharp. If at all rounded they will allow the tooth to escape too soon.

(7) The pivots can only be allowed a very slight amount of play in the pivot-holes; for a minute change in the relative positions of the mobiles will produce a considerable variation in the lift, etc. It is needless to point out that, with relatively too small holes, the thickening of oil will interfere with and modify the motion of the several parts.

(8) Even estimating the pitch of the tooth in the notch at one-twelfth the diameter of the roller, although it is less, it follows that this amount of pitch with a roller whose diameter is one-sixth of the distance between the points of two successive teeth will be less than a twenty-fourth part of the diameter of a roller of one-third, a quantity so minute that we may feel sure that some slight want of truth in the roller or an

imperceptible wear at the points, etc., will cause the escapement to fail in its action.

(9) If, in order to avoid the inconvenience which thus results from too shallow a pitch, it be made deeper by bringing the mobiles nearer together, the risk of setting is increased and the shallowness of the notch also occasions some difficulty. If a tooth pitch to the same actual depth in two rollers, that measure respectively two-sixths and one-sixth, this tooth will give a small lift in the latter case nearly a third as much again as that in the former case; it follows, then, that the balance will traverse an arc of greater extent before the tooth is free.

In short, when the roller is very small the action of the escapement must take place *with a mathematical exactness that is extremely difficult to attain and which it is absolutely impossible to guarantee.*

Experimental Data.

529.—The watchmaker must always keep clearly in mind the fact that, in the great majority of cases, he will not obtain a satisfactory result by diminishing the roller beyond what is prudent but by modifying the other portions of the escapement.

At the present day practical men, taught by long experience, are agreed in considering that the diameter of the roller should in no case be reduced below two-eighths, that is one-fourth, the distance between the points of two teeth; and even this amount they will only employ in the smaller callipers of watches. In the larger and thicker class they consider that the roller should be two-sixths or one-third of that interval in order to avoid bending of the staff and vibration of the axis. English watches are to be met with in which it is as much as two-fifths, but this is too great, for it must not be forgotten that, as the roller is increased, the locking action is less safe, since an arc of a circle approximates more and more to a straight line as its radius of curvature becomes greater.

The average diameter of the roller should be intermediate between two-eighths and two-sixths, that is to say two-sevenths, or a very little in excess of this (fig. 17, plate V.). Such a size is suitable for most well-made French watches; and we would here point out that the use of the duplex escapement, requiring as it does a considerable motive force and some thickness, should rather be confined to watches of medium size. The motive force is deficient in small thin ones, while in those

which are large and thick, such as the English, the weight of the several parts and the great motive force make the untangential rest the more detrimental and occasion rapid wear of the pivots.

Great Lift.—Length of Impulse Pallet.

530.—It may be laid down as a general rule that the arrangement of every escapement must depend on the extent of the lift; and this lifting angle is itself a consequence of (1) the intensity of the motive force and (2) the relative velocities of the several mobiles that act on one another.

Experience has indicated that, in the majority of cases, a great lift of 30° (including the drop) is sufficient. When such an angle is found not to produce a movement of the balance of the requisite extent, the cause of the deficiency must be sought in either a weak mainspring or a faulty construction of the mechanism.

If the motive force is not excessive, any increase in the lift will fail to increase the extent of motion of the balance altogether, or will only do so in a very slight degree. The force is merely resolved in a more oblique direction, and the only effect is a greater lateral pressure on the pivots.

531.—The length of impulse pallet, with a given interval between the centres, cannot be arbitrarily chosen, but is a definite function of the size of the impulse wheel and the number of its teeth. Clearly it can be longer as the interval between two teeth is greater, a fact which is indicated by fig. 22, plate V., where the pallet *a c* would be reduced to *a b* with thrice the number of teeth.

It follows that the lift applied to the pallet will be greater as the number of teeth is reduced.

Assuming the two centres of motion F and N (fig. 16, plate V.) to remain fixed, if the size of wheel be varied, its radius in the first case being F D, and in the second half of this amount or F O, the interval between the teeth will in this second case be half what it was at first, for the arc A D B is double the arc C O H (the letters A, B, H, C indicate the positions of the four teeth). Further, it will be observed that the two circles A X B, C Z H, described by the extremities of the pallets, do not overlap the rim of the two wheels to the same extent, for the distance X D is at least four times as great as O Z; and the lifting angles differ in a similar manner. The lift with the shorter pallet is measured by the angle A N B and, with the

longer, it is represented by C N H, an angle which cannot be increased owing to the danger of the teeth catching.

532.—This geometrical demonstration proves that:

(1) We cannot directly increase the *great lift* at will, but it varies inversely with the length of the pallet, that is to say it increases as we diminish this length providing the small wheel is proportionately increased.

(2) The more an impulse wheel is increased and an impulse pallet decreased, the deeper will be the pitch between the tooth and this pallet, and it will thus be the more safe; whereas when the wheel is reduced and the pallet made longer the contrary is the case.

Any watchmaker will at once see the inconveniences of such extremes; and he will conclude that, between a very short lift, where the tooth little more than touches the pallet, and an excessive lift which compels the balance to traverse an arc of considerable extent before it can escape, the best proportion is at some intermediate point that must be ascertained by experiment. As a matter of fact, experience has shown that this point lies somewhere between a lift of 30° and 45°. The absolute length of the impulse pallet cannot then be fixed upon until the amount of lift, the number of teeth of the wheel and even the size of the wheel have been settled. For, if we retain the *small lift* the same while diminishing the roller and bringing the centres of movement proportionately nearer together, the two impulse teeth *j* and *h* (fig. 12, plate V.) will be held at a rather greater distance from the line of centres E B, the one *h* during the rest and the other *j* at the end of the small lift; this being the case it will become necessary to somewhat reduce the pallet in length.

Anticipating what follows we may state that when the great lift is about 35°, the wheel 15-toothed and the roller two-sevenths the interval between the points of two of the teeth, the length of pallet, measuring from the centre of the staff, is just over three-eighths the radius of the locking wheel.

Ratio between the diameters of the two wheels.

533.—It is a mistake to assert as an unvarying rule that the ratio 3 to 2 is better than 4 to 3 or conversely.

The proportion for the locking wheel to bear to the impulse wheel cannot be decided upon *a priori;* it follows as a necessary consequence of the amount of lift, for the diameter of the latter

wheel must gradually increase with an increase of the lifting angle. This is evident from an inspection of figures 1 and 3 of plate II. In fig. 3, where the lift is only 35°, the wheels are to each other as 3 is to 2. Whereas in fig. 1, with a lifting angle of 48°, the ratio is about 4 to 3.

534.—The early watchmakers who studied the duplex escapement endeavoured to arrive at the true principle of its action by a line of argument somewhat as follows:—"The laws of Mechanics teach us that, in overcoming any resistance by means of a lever, the force available will become greater as we increase the length of the arm at the extremity of which the power is applied. (See the articles on the *Lever* in the Introduction, page 22.) In other words, when two lever arms act in conjunction, the power is to the resistance in the inverse ratio of the lengths of those arms.

"Now, observing that the escapement pallet is no more than an arm (a resistance or inactive arm) on which the radius of the wheel acts as a power or active arm, it is evident, applying the above principle, that the pallet must be prolonged as much as possible, the impulse wheel being proportionately reduced in diameter; and thus, comparing it with the locking wheel, the difference between their two diameters must be as great as possible."

535.—This conclusion, like the one from which they sought to deduce the best size for the roller (**526**), is false, for the above reasoning ignores many important secondary conditions, and thus the complexity of the action with which the problem has to deal is left out of account.

When studying the action of the several parts of an escapement, whether they be levers or inclined planes, we must carefully guard against the error of considering them only at the instant the balance commences its motion, an imperceptible fraction of the total period of the vibration and so short that the velocities and momenta acquired during the interval are almost nothing. The escapement must, so to speak, be regarded when in the full performance of its several functions, that is, beating its full number of vibrations per hour; 18,000 for example.

536.—This being granted, consider a power lever, at first measuring 2, and subsequently reduced to 1, the contrary being the case with the resistance arm; when the active lever is reduced to half its initial length, the force exerted at its ex-

tremity will be doubled in intensity but this extremity will only traverse a space of half the amount. Hence the passive or resistance lever is, in the second case, only moved through half the space which it traverses in the same period in the first case (assuming that these periods are equal although this is not absolutely true); it follows that with the longer power arm the resistance lever is driven by a force 1 through a space 2, while with the short power arm the other moves under the influence of a force 2 through a space 1.

As the impulse applied to the pallet is a product of the force acting on the wheel multiplied by its velocity, the resulting action is clearly the same in the two cases, and it appears to follow, at first sight, that it is a matter of indifference whether we increase or decrease the relative lengths of the arms.

537.—This conclusion, however, although apparently legitimate is far from the truth. The question we have to consider is not a simple one but complex and before drawing any conclusions we must give due weight to the three following important facts:

Firstly, the angular velocity of the wheel as compared with that of the balance increases as we lengthen the pallet, the radius of the wheel being at the same time proportionately reduced.*

Secondly, the period during which the tooth acts on the pallet becomes less as the radius of the wheel is diminished, the length of pallet being proportionately increased (**532**—1).

Thirdly, the pitch or extent to which the mobiles are engaged increases as the pallet is shortened and the radius of the wheel made greater (**532**—2).

538.—From these observations the following consequences naturally follow: *with a very long impulse pallet*, and therefore a small impulse wheel, in addition to the danger of the escapement failing in its action, the tooth escapes from the pallet almost immediately after it has engaged with it. The

* *Demonstration.*—Let the length of impulse pallet be B C (fig. 11, plate V.), and the radius of the wheel double this amount, or A C, the angle D B C being 40°; while the balance traverses the angle D B C, the wheel will move through H A C. Since A C is the double of B C it follows that the angle H A C is half of D B C, that is, it is 20°. The balance will therefore move twice as fast as the wheel.

If the radii of the pallet and wheel B M and A M are equal, the angle B A M will, like A B M, be 40° and the wheel and balance will move with equal velocities.

It will thus be seen that the rate at which the wheel moves continually increases and approximates to that of the balance when the wheel is diminished, the pallet being proportionately lengthened.

lift is reduced to little more than a rather prolonged drop. The force applied to the balance is almost nothing because the wheel travels quite as rapidly as it does, and the only effect of any excess of velocity in the wheel is to increase the force of impact on the locking surface; an impact that is the more detrimental from being accompanied by engaging friction. The retarding force (**318**) is in excess, and it is obvious that to increase the accelerating force at the expense of the other it is only necessary to shorten the pallet and proportionately increase the size of the wheel. It thus appears that to magnify the force of the impulse it will suffice to make the pallet shorter and shorter. Although this is the case it is only true between certain limits. When *the pallet becomes very short* and the impulse wheel large, the drop is very much deadened since the wheel travels slower than the balance; but such an advantage is inconsiderable as compared with its accompanying disadvantages, for the diminished velocity of the wheel makes it simply follow after the pallet as it were without applying any efficient push and, besides this, a great part of the action of the wheel on the pallet takes place in a direction very oblique to the line of centres. It thus happens that the force is decomposed in such a manner that a considerable portion is detrimental to the action of the escapement, for it is utterly lost for the purpose of maintaining motion and only results in an increase of friction. (See the Introduction to the study of Escapements, articles on *Lift*, etc.)

539.—We clearly see then that there must be some mean value, mainly dependent on the motive force, at which the advantages and inconveniences of the two extremes meet, where the accelerating and retarding forces balance, so that neither of them, by being in excess, is liable to interfere with the action of the regulator, and exercise any injurious effect on its timing qualities.

As was the case with the height of incline in the cylinder escapement, this mean must be determined experimentally; it will be the point at which the oscillation is of the requisite amplitude, the lifting arc being as small as possible.

By satisfying such a condition the following advantages are secured:

(1) The pitch of the impulse tooth and pallet is such as to ensure their proper action;

(2) The two remain in contact through an arc of about 30°, which is the great lift of the escapement, and experience goes to prove that such a great lift is sufficient with a duplex providing its thickness is well proportioned (**97**).

(3) The action of the tooth takes place very approximately in the line of centres and perpendicular to that line; this is an advantage (**73**).

(4) As the motion of the wheel while impelling the balance is very nearly uniform, the drop is less violent than if the movement of the wheel were accelerated in a manner similar to that of the balance.

Experimental Data.

540.—It may be accepted then as a fairly general rule that, for watches of sufficient thickness, the proportion 3 to 2 is best. A lift of about 35° is the result.

As the extreme limit for the length of impulse pallet in such watches is fixed by a lift of 30°, it is clear from what precedes that with less motive force the lift must be a few degrees greater. The balance, naturally lighter, will, in consequence of its less mass, be set in motion more easily; but since from this same reason it is less capable of continuing its motion (**323**), the impulse to which it is due must be of longer duration or more frequent (**325** and **326**). This fact will be the more obvious when it is remembered that the duplex wheel is heavy and becomes more sluggish as the motive force is reduced; its action at the commencement of the lift will thus become gradually less effectual.

541.—Hence in thin watches or such as have a weak motive force the great lift may be increased. If the diameter of the impulse wheel be increased to three-fourths that of the locking wheel, we obtain a lift of from 45° to 50°; but with such a lifting angle, if the balance is at all small and heavy and the motive force deficient, the disadvantages of a short pallet soon become conspicuous; if, on the other hand, the balance is at all light and the motive force great, we are met by the difficulty of vibrations too much extended and are thus no better off.

To sum up, then, the less 40° of lift (inclusive of the drop) is exceeded as a means of fixing the dimensions of the impulse wheel the better, except of course where it is impossible to avoid going beyond it.

Position of the Impulse Tooth.

542.—The position occupied by an impulse tooth between two locking teeth cannot be fixed upon arbitrarily; it must always be set midway between those two teeth.

For during the dumb vibration the pallet passes near the right-hand impulse tooth *h* (fig. 12, plate V.) whereas on the return of the balance, in the following vibration, it grazes past the left-hand tooth *j*; but this only occurs at the end of the lift, when the locking tooth, having traversed the arc *a c*, is on the point of escaping from the roller notch.

As the impulse tooth *j* will traverse the arc *j n* while the locking tooth moves through *a c*, it is evident that, if we consider the point of the tooth *o* to be arrested in the middle of the lift, it will rest on the line of centres E B and the pallet will pass at equal distances from *h* and *j*.

The interval *o h* must equal *o j*; but *o j* is the same as *d h* if the wheel is accurately divided as it should be. Thus the point *h* is shown to be midway between the lines *o a* and *d k*.

Small Lift.—Pitch of the Locking Tooth in the Roller Notch.

543.—In addition to committing themselves to many other contradictions, numerous authors and practical men fix the pitching of the locking wheel in the notch at one-sixth the diameter of the roller, notwithstanding that they vary the small lift in the proportion of 1 to 2. It is impossible to reconcile such practices, for every increase in the small lift must necessarily correspond to a more near approach of the point of the tooth to the centre of the roller, and *vice versâ*. It is inconvenient to express this depth as a fraction of the diameter of the roller, for its amount depends on the lift, the width of notch and the rounding of its edges.

When the small lift is 30° the pitch is something less than a sixth, an amount which seems so minute, especially in the smaller class of watches, that one would fear lest the locking tooth should fail in its action if we were not aware that this is not the point in which the duplex usually goes wrong.

Those who recommend a small lift of 50° as a datum for fixing this depth have overlooked the fact that the rounding of the edges, often very marked, causes a loss of some degrees in the lift, and that to attain to the requisite amount the mobiles must always be rather closer together than the drawings

indicate. As a matter of fact, in practice the tooth always pitches in the notch to a depth slightly greater than that indicated in most drawings of escapements.

As experience has proved a small lift of 30° (as a maximum) sufficient to ensure the proper performance of the locking action, it is useless to exceed this amount, for by doing so we should only increase the risk of setting (**528**—9).

The Roller Notch.

544.—The notch must be of such a width as to allow of the free action of the tooth during the performance of an entire small lift. In every position of this tooth during its passage a slight amount of freedom is essential. It is impossible to accurately define the best size for the notch as it must depend on the pitch, form and thickness of the teeth; the two latter data vary with different wheels as may be thought necessary in accordance with the metal employed, whether it be brass, steel or English metal; the teeth being made thicker or thinner at their points as the metal is more or less liable to distortion, wear, or flexure.

The edges or lips of the notch must only be very slightly rounded and, like the surface of the roller, must be highly polished.

Some authors assert that the notch should measure 30° and the rounding of the lips 10°, and at the same time they prescribe a small lift of 20°. Others think this to be a mistake, since the angle of the opening (C D, fig. 15, plate V.) is greater than the angle of lift, A B, and therefore, they say, the tooth will pass by the notch without touching the roller.

In the article which treats of the designing of a duplex escapement (**548**) the method to be adopted in determining the width and size of the notch will be explained.

On the First or Impulse Drop.

545.—The extent of the drop which occurs between the small and great lift should be, according to some authors and practical men, 4°; according to others, 10° or more than double the previous amount. The first class, being aware of the disadvantages involved in extensive drops, are doubtless anxious to limit their amount as much as possible. But this is quite different to the ordinary case, for it must be remembered that when the drop is 4° the pitch of the pallet with the impulse teeth is very shallow; its engagement is therefore uncertain and

occurs to a great extent at the corner of the pallet, being accompanied by engaging friction.

A drop of 10° materially reduces these two faults and causes the action to take place more nearly in the line of centres and perpendicular to that line; these are important advantages which more than compensate for the drop having somewhat greater energy, a comparatively unimportant fault.

We are here discussing the case of a watch beating 18,000 vibrations per hour and of the form usually adopted at the present day, for, as has been already explained, the drop must vary with the weight of the wheel, the motive force and the number of vibrations.

Some practical men arrange that the tooth and pallet only come in contact on reaching the line of centres and they assert that there is no inconvenience experienced on doing so; but in such a case the severity of the impact must occasion a considerable shake and the great lift is reduced to nearly one half.

RÉSUMÉ.

546.—The great majority of modern duplex watches that have a good rate are proportioned as follows:

Size of roller:—two-sevenths the distance between two points of teeth.

Great lift:—between 30° and 35°.

Small ,, ,, 20° and 30°.

It has been already shown that a great lift of 30° or 35° fixes the diameter of the impulse wheel at two-thirds that of the locking wheel or slightly more.

Drop between the two lifts:—10°.

Width of the notch:—enough to permit of the passage of the tooth or rather a little more.

Length of impulse pallet measured from the centre of balance-staff:—just over three-eighths the radius of the locking wheel.

It is better that the pallet be at first rather longer than the amount here indicated, whether it be made entirely of steel or provided with a brass facing to serve as a model to the lapidary; it can be shortened as necessary and the escapement verified in position on the depthing tool.

Remarks.

547.—The above dimensions are suitable for French watches of sufficient thickness and of gentleman's size. They

can also be adopted in large thick watches providing the size of the roller is increased progressively to about one-third of the interval between two teeth.

In watches with a weak motive force and a consequently light balance (thin watches for example), the roller may be reduced to a quarter of this interval and the diameter of the impulse wheel made three-quarters instead of two-thirds that of the resting wheel; that is, it will increase according as the great lift is desired to approximate to 45° or 50°. At the same time it is best not to use this form of escapement in watches of such dimensions.

Two circumstances will enable us to ascertain whether the several parts are properly harmonized: the timing and the rate maintained when varying motive forces are applied.

TO DESIGN A DUPLEX ESCAPEMENT.

548.—The following method should be adopted in drawing a duplex escapement, the principal dimensions of which we will assume are as follows: wheel 10 mm. (0·394 ins.) in diameter and provided with 15 teeth;—diameter of roller two-sevenths;—great lift, including drop, 35°; small lift, 30°.

The known diameter of the wheel is multiplied by 20, 30 or 40, etc., and the drawing will then be such that any dimension of the actual escapement can be deduced by dividing by one of these figures.

Multiplying the diameter, 10 mm., by 30 we obtain 30 centimetres (11·811 ins.) as the diameter of the wheel, and 15 cm. (5·905 ins.) as its radius.

On a sheet of good drawing paper fixed on a board draw the line of centres A B G (fig. 3, plate II.).

With the centre A and a radius of 15 cm. (5·905 ins.) describe the circular arc E K H L which gives the size of the wheel.

As the entire circumference measures 360°, by dividing this amount by 15 we shall obtain the interval between two teeth (24°).

By the aid of a protractor or by one of the more accurate methods subsequently described, measure off the angle K A H, of 24°, such that the line of centres divides it into two equal parts; the angle K A B is thus equal to B A H.

The interval between the points of two successive teeth K H must be divided (by a compass or protractor) into 6, 7 or 8 equal parts according as the roller is required to measure one-third, two-sevenths or one quarter. In the present case 7 divisions are taken, since the roller is to measure two-sevenths.

Describe on a small piece of cardboard a circle with a radius equal to one of these divisions; it will thus represent the roller. Mark off an angle, *a b c*, of 30° on the circle (by mistake made 40° in the annexed fig. 39), and bisect it by the line *x z*. Now cut out this roller and lay it on fig. 3 (plate II.) between B and G.

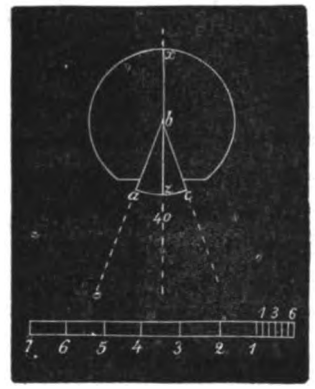

Fig. 39.

Slide it gradually down from G towards B (taking care that the line *x z* coincides accurately with G A) until the two extreme points *a* and *c* of the angle *a b c* (fig. 39) meet the circumference of the wheel (E K H L). When in this position mark with a compass point the centre of the roller B *b*, and the two points *d* and *c* immediately under the points *a* and *c* of the cardboard disc; this disc may now be removed as useless.

With the centre B (fig. 3, plate II.) describe a circle to represent the roller and draw the lines B *d* Z, B *c* X. If the drawing has been made with care, the angle Z B X will equal 30°, and the circumference of the roller, as well as the two lines B Z, B X, will cut the circumference of the wheel at the same points *d*, *c*.*

The rounding of the lips measures about 5° on either side; if 5°, then, be marked off to the right and left of Z B X we shall have the angle marked 40° in the figure. Its left-hand bounding line indicates the right-hand side of the notch, the rounding of which extends to the line B Z.

By drawing the lines A *c*, A *d* from the centre of the wheel

* Most designers of escapements adopt a different plan to ascertain the position occupied by the centre of the roller. Having drawn a straight line whose length is equal to the interval between two teeth (fig. 39), they divide it into six, seven, or eight equal parts, one of which is further subdivided into six equal parts; as the tooth is assumed to be pitched in the roller to the extent of one-sixth of its radius, five of these subdivisions represent the distance between the line *d c* and B (fig. 3, plate II.), or the centre of the roller. Besides the difficulty experienced in accurately dividing the scale, this method is objectionable since it gives an invariable small lift which can never be made less than 50°.

to the points *c* and *d* we obtain the angle *d* A *c*, and this gives a measure of the space through which the escape-wheel moves during the entire small lift.

The roller notch occupies a space of about 20° in the drawing; a glance will suffice to show that this is more than is required to ensure safety of action, even although the centres of motion be brought slightly nearer to each other. The amount of the opening should not reach 30° except in the case of escapements in which the small lift is more than 50°.

We thus now know:

The two *planting points* or centres of rotation of the wheel and roller,—the size of resting wheel,—size of roller and width of its notch,—the angular measure of the small lift or the displacement of the roller,—the angular path traversed by the wheel during an entire small lift.

So there still remain to be determined:

The angle of great lift,—the length and position of impulse pallet,—the size of impulse wheel and position of its teeth,—the positions of the resting teeth.

Since *d* is the point at which rest occurs on the roller, the line *d* A will fix the position of the tooth *dj*. Draw the two lines A E and A L inclined at angles of 24° on either side of A *j d*. As this amount is the interval between two consecutive teeth, the resting teeth will lie on the lines A E, A *j d*, A L, and they can be drawn in afterwards.

The impulse tooth must be midway between the points of each pair of resting teeth, so draw the lines A *f*, A *g*, equally subdividing the angles E A *d*, *d* A L. They will subsequently enable us to ascertain the positions of the two impulse teeth, since they must always be placed on these division lines.

From the centre of the roller draw the lines B *m*, B *n* inclined to each other at an angle of 35° (the great lift) which is bisected by the line of centres; the angle A B *n* thus measures 17·5°.

The point V, where the lines B *n*, A *g*, cross, indicates: (1) the position of the impulse tooth; (2) the length of the pallet; (3) the size of the small or impulse wheel.

From the centre of the roller and with the radius B V describe the circumference Q P V traced out by the pallet. With the centre A and radius A V draw the circle R S V T representing the size of the impulse wheel.

The resting teeth, E, *d j*, L, can now be inked in, as also the smaller teeth R and V.

Measure off on the rim of the wheel the distance *i j*, that is the interval between the two lines *d* A, *c* A. This distance is set off at R S and V T, and thus gives the space through which the impulse teeth are moved forward during the small lift. For the wheel advances by this amount and the teeth will therefore be, at its termination, at S and T, as shown in dotted lines.

The point P, or the position of the pallet at the termination of a small lift, is obtained by measuring off an arc S P of 10° to represent the drop; then from P set off the arc P Q equal to *z x*, that is, the small lifting arc, and the point Q, thus determined, gives the position of the pallet at the commencement of a small lift. (By an error of the engraver this point is placed too much to the left.)

The several proportions of the escapement have now been secured.

If we consider the escapement to be in action, it is evident that, while the rest is occurring at the point *d*, the dumb vibration is taking place and the pallet will just pass by the tooth V without contact; in returning, the pallet will be in the position Q when the small lift (of 30°) commences, and during this lift will traverse the arc Q P, also equal to 30°.

During the latter movement the pallet will travel quicker than the tooth R and will thus pass in front of it with a slight freedom; the tooth will not reach S until the pallet is at P.

The point of the resting tooth in contact with the roller being now at *c* will escape from the notch, and the impulse tooth S will, after a drop of 10°, fall on the impulse pallet in the position shown by the dotted line P *j*, and, impelling it through the entire arc P V, will produce the great lift which is immediately followed by the resting of the tooth E against the roller.

Remarks.

The great lift, occupying 35° in the drawing, is in reality only about 20° or 22°, since the 10°, S B P, are required for the drop, and from 3° to 5° are lost through the rounding of the corner Q of the pallet. If it be required that the tooth should impel the pallet through an arc of 30°, we must add to this amount 10° on account of the drop and a few degrees for the rounding of the corner Q; the angle *m* B *n* will thus become 40° or 45°.

The drop however, in this case, does not quite amount to 10°: the rounding of the corner of the notch somewhat reduces it. But we have neglected this slight difference in order to avoid confusion.

It will of course be evident that the drawing cannot indicate

the plays and freedoms with any certainty. Such microscopic quantities can only be accurately ascertained when verifying the escapement on the depthing-tool.

CHAPTER III.

PRACTICAL DETAILS OF CONSTRUCTION.

To Trace out the Calliper of the Escapement.

549.—This is merely an abridgment of the method explained in the preceding article.

If the escape-wheel is already completely made it will be useless to trace out the calliper of the escapement. The roller and impulse pallet are made in accordance with the dimensions of this wheel and corrected, if necessary, after the verification on the depthing-tool.

We will consider the case in which nothing is known but the distance apart of the two centres of movement and the number of teeth of the wheel.

Take two points on the surface of a smooth sheet of brass to indicate the centres of the wheel and roller respectively. Join them by a fine line, the line of centres (*a b*, fig. 14, plate V.).

If we assume the wheel to have 15 teeth, their interval apart will be 24° ($24 \times 15 = 360$). From the centre of the wheel (*a*, fig. 14) draw two lines inclined at an angle of 12° on either side of the line of centres, thus enclosing a total angle of 24°.

From the centre of the roller, *b*, also draw two lines enclosing an angle equal to the great lift (inclusive of the drop and the interval of safety), that is 35°, which must be accurately bisected by the line of centres.

From the centre *a* draw the arc *d d'* just beyond the points of intersection of the four lines drawn as above directed; this circumference gives the diameter of the impulse wheel.

The drawing can now be made on a larger scale on paper or cardboard, and, since the diameter of the roller is determined by the interval between the resting teeth, which interval is given by the two lines inclined at an angle of 24°, it will be easy, by trial, to soon find out the requisite size of roller and the length of resting teeth. These data having been ascertained, they are incorporated in the smaller drawing, which thus becomes a complete calliper of the escapement.

Remarks.

The protractor of a mathematical instrument box is very unsatisfactory as a means of ascertaining these angles accurately; a better means is to employ the divisions of a wheel-cutting engine.

It will be as well to make the resting wheel rather larger than necessary, and, after verification in the depthing-tool, this excess can be removed when trueing the wheel (**553**).

Details concerning the Duplex Wheel.

550.—The wheel may be made either of steel or brass. A steel wheel, owing to its greater rigidity, can be more easily made true, but it is objectionable from the fact that the friction of steel against rubies (other than the Oriental) causes the formation of an oxide when the oil is at all dry or decomposed; this oxide, under the pressure of such harsh engaging friction as that which occurs during the rest in this escapement, is detrimental not only to the points of the teeth but even to the ruby roller, and this loses its polish and sometimes even becomes pitted, an accident which is not uncommon in English watches where the motive force is excessive.

Although such a disadvantage is not possessed by a brass wheel it has another almost as serious, namely the difficulty experienced in cutting it. Brass is always distorted by a kind of molecular change set up by the impacts and pressures that are unavoidable in cutting a wheel. Hence it is exceedingly difficult to form accurately teeth that are so fine as those of the resting wheel.

The best wheels that have been made up to the present were made in England; they are beautifully finished; the methods and tools, however, employed in their construction are kept secret by the workmen in whose possession they are. The metal of which they are formed is an alloy resembling well hammered brass, but better able to resist distortion when the wheel is being cut. It is known in France as English brass (*laiton anglais*).

The resting teeth should be thin and tapered (**91**) and they may be cut either with the front face radial or may be star-like. Some makers incline them a little backwards or else make a short inclined plane on the front side of the point, which rubs against the roller. Such a precaution reduces the recoil of the wheel when the motion of the balance is reversed, but there is danger of a butting action occasioned by the bending of the staff.

The impulse teeth should be triangular prisms with their faces inclined forwards as seen in fig. 3, plate II. In fact they should resemble the teeth of what is known as a *ratchet* wheel.

The face should be inclined at an angle of about 17° to the radius, R A, of the wheel.

There is found to be a practical advantage in making this inclination such that, when the tooth engages with the pallet (R′ *z*″, fig. 1, plate II.), the two faces in contact are approximately parallel. Such an arrangement in no way interferes with the timing and is conducive to the preservation of the surfaces, for when the impulse teeth are too much inclined they are found to wear at the points.

The practice, much in vogue in England, of employing, in most forms of escapement, small escape-wheels and heavy balances that are somewhat large and perform long vibrations, is less suited to the cylinder and duplex than to other escapements. In the first a small wheel involves a too small cylinder, and the inconveniences of this have already been indicated. In the duplex the wheel had better be too large than too small. The friction of the rest will be less severe and the proper action of the pallet will be more assured, owing to the interval between the impulse teeth being greater.

In factories the size or rather the radius of the wheel is determined by the fourth wheel of the train, an interval of safety being allowed between the points of the resting teeth and the fourth wheel pinion.

Such an empirical rule is, as we have already observed, useless, and we would refer the reader to article **399**; the slight restriction mentioned in the last paragraph must however be taken into account.

The duplex wheel has considerable weight owing to its peculiar form and double set of teeth; it is well, therefore, to make it work between endstones, as then the friction is reduced and less delay occurs in the commencement of its motion.

When the wheel is of steel it is a matter of indifference whether it be hardened or not.

The surface of the roller and the notch should be liberally supplied with oil, and a little must also be placed on the points of the resting teeth. The pallet and impulse teeth also, when they are of steel, require oil. It is necessary with a ruby pallet

and steel wheel, but need not be applied with a brass wheel and ruby pallet; some watchmakers maintain, however, that even in this case the application of oil is beneficial.

A wheel made of English brass acting in conjunction with a simple steel pallet preserves its surface very well without using oil. Indeed we have noticed cases in which its presence has actually caused the pallet to lose its polish with some rapidity.

To make a duplex wheel.

551.—If not provided with a special tool a workman will require a well-made wheel-cutting engine, accurately finished in all its parts, in order to make a duplex wheel; for we would again point out that such a wheel offers real difficulties of construction, and a watchmaker who is not skilled in this kind of work will experience some difficulty at first.

Makers of escapements on a large scale have a special series of tools applicable to any particular form of wheel, etc.; but it is not so much peculiar methods of procedure as their prolonged practice in the making of one particular thing that gives them their superiority. It will of course be useless for us to do more than describe the methods to be adopted when employing the tools that every watchmaker may be supposed to possess.

The steel of which it is intended to form a wheel must be prepared as already indicated for the cylinder escape-wheel (**484**).

When it is formed of brass, this metal must be of the very best quality and its thickness must be reduced by careful hammering to but little more than that of the finished wheel. Whether it be of steel or brass it is essential that the turning be accomplished by means of sharp cutters, so as to avoid all necessity of applying an excessive pressure.

On a smooth plate of brass a circle is drawn giving the tota size of the resting wheel, and within this two others, that indicate the size and width of the rim of the impulse wheel (c, fig. 14, plate V.).

When the wheel has been turned down to the correct diameter and the two faces are parallel, the flange for forming the resting teeth is cut and the interior hollowed out in a small lathe with a slide-rest or, in its absence, in ordinary turns employing hooked gravers (**484**).

Any watchmaker will easily see by mere inspection what should be the height of the impulse teeth and thickness of the wheel; this must be light but at the same time rigid.

Having turned the wheel up true and of the right size and thickness, set out three arms and smoothed both faces, it is firmly fixed by shellac or otherwise to the table of the wheel-cutting engine. This table usually carries at its centre a perfectly true rod which the hole in the middle of the wheel fits easily but without play.

The cutter employed must not remove the entire mass of metal between two teeth at once; it would be utterly impossible to cut the teeth true by such means. Its thickness should be rather more than one-third the interval between two teeth, and the metal must be removed by three operations, thus rendering necessary three complete rotations of the table in cutting the resting teeth and the same number to form the impulse teeth.

It is well to first form the inclined faces of the great teeth, and this inclination is obtained by setting the cutter somewhat to one side (fig. 19, plate V.). When this first turn of the table is completed, the cutter is brought opposite to the centre of the wheel and by an adjusting screw it is set as indicated in fig. 20; the middle portion of the space is thus cleared. When this series is complete, the wheel is again moved so that it takes up the position shown in fig. 21, and the matter contiguous to the straight faces of the teeth is removed.

The two faces of the impulse teeth are then formed in a similar manner.

After the wheel has been taken from the table, all burrs produced by the cutter are removed, the corners are rounded and the points of the resting teeth and faces of impulse teeth are polished with care.

The corners of the small teeth must be sharply defined although delicately rounded. The same is the case with the extremities of the resting teeth which should be thin but not sharp. These latter are slightly rounded on their sides.

Escapement-makers remove the marks of the cutter by polishing (with a lap charged with rouge) the faces of the teeth and the rim of the wheel between them in a special tool, the supporting platform of which has a circular oscillating motion dependent on the length of a space.

When such a tool is not accessible this operation can be done by hand, as it only requires care and a light touch.

552.—Remarks.—A 15-tooth wheel can be cut by using the 90 ring of the divided plate, taking every sixth division

round the circumference and then passing over one division after each complete rotation of the plate. When, after the third turn, the cutter has reached the faces of the resting teeth, it will be only necessary to set the index in the third point in advance to cut the front straight faces of the impulse teeth, that is to say these faces will be exactly midway between the straight faces of the resting teeth.

The teeth are cut with a single rotating cutter or a plain circular cutter which is only roughed on its edge, the two faces being slightly dished, but the corners must be sharp. Very little metal must be removed at a time. The single cutter is only capable of making a clean cut when the tool is very firm and the velocity of rotation considerable. Such an appliance as that represented in fig. 18, plate V., is often used; it partly resembles both forms of cutter and should be very hard and polished on the sloping faces.

Or we may adopt the method recommended for the cutting of detent escape-wheels.

Whatever tool be used it must be first tested on a trial wheel; at the same time the several positions to be occupied by the cutter can be ascertained. In order to be able to restore it to these positions when necessary, they should be recorded by notches made with the cutter itself in a brass rule which fits between the two fixed centres that carry the arbor and cutter.

When not formed of steel the wheel may be strengthened by a supporting brass disc.

To make the escape-wheel true and to remove burrs.

553.—These two operations require to be performed with extreme delicacy when teeth are so long and far apart as those of a duplex wheel. The practice of topping the wheel or removing the burrs by simply holding a fine flat file on the rest of the ordinary turns, which is adopted by many watchmakers, nearly always results in a distortion of the wheel and there is some danger of straining the teeth.

The following is a good method of performing the operation.

The wheel, placed on an arbor, is supported between two centres of the depthing tool; the axis of these is at *a*, fig. 23, plate V.

The disc *d c f*, provided with a shoulder, is put on one of the other pair of centres *(b)* opposite to the wheel. It carries

a small ruby file or a very hard oilstone cemented in the position *d c*, and the whole is maintained firm by the clamping screw *f*.

Rotating the wheel by a very light hair bow, the adjusting screw of the tool is moved until the teeth just touch the surface of the stone. The depthing tool is now laid on its side on the bench so that it is supported by the edge of its base and the heads of the two thumbscrews, and one hand works the bow while two fingers of the other lightly press upon the uppermost branch of the tool: this slight force is sufficient to make the stone act on the points of the teeth. Should it have failed to touch some of the teeth they must be again brought into contact by the screw and the operation repeated with the greatest possible care.

The burrs can also be removed on the ordinary finishing turns if the rest be replaced by a special support carrying a small adjustable frame that can be inclined as required by means of a micrometer screw. The frame carries a very fine flat file which can be safely brought, by the screw, in contact with the points of the teeth.

The thread of the screw should be tight-fitting and of very slight pitch.

The tool used in equalizing the teeth of a verge wheel may be employed for making those of a duplex wheel true. The curved piece is replaced by a straight arm, and against this each impulse tooth is supported in turn; a cutter is then brought against each resting tooth, and by this means the entire wheel will be equalized.

To make a Balance-Staff.

554.—The construction of the balance-staff presents no difficulty and is similar to that of the seconds wheel pivot (**567**).

Pivots and Pivot Holes.

555.—It is extremely important that the pivots of a duplex escapement all work in jewelled holes (Oriental rubies or sapphires) so that the positions of the several mobiles may remain invariable; the inside of the holes must be very highly polished; for, since the pressure is considerable, the pivots are driven with some force against the sides of their holes, and any roughness in these latter will occasion a more rapid wear.

With the same object in view the oil-cups of these jewels should be deep rather than wide, the space between the jewel and endstone almost imperceptible, and the surface of the stone

rounded in tallow-drop form. In short, everything should be done that experience has shown to be conducive to the keeping of oil between rubbing surfaces (**81**).

Special care must be taken to ensure that the pivots are hard, highly polished and perfectly round. As the staff in a duplex escapement is required to carry a balance heavier than that employed in a cylinder watch, it is advisable to make the pivots of the form indicated at *a* A, fig 40, so that they may be fine and yet strong; they should be rather long in order that the oil may not be drawn toward the body of the axis.

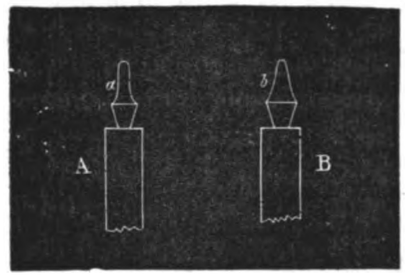

Fig. 40.

The majority of the pivots of this form that are to be met with in watches are badly made, often too short and, with few exceptions, too much coned or of a doubly conical shape. Hence it follows that, if the watch is inverted, the escapement is no longer under the same conditions (as to friction) as before.

That portion of the pivot which works in the jewel *(a)* should be truly cylindrical.

We shall refer again to this subject when considering *Pivots* in connection with the Lever Escapement and will, therefore, only observe that a truly conical pivot (*b*, fig. 40), known also as an *English pivot*, is very useful in facilitating the timing in position, but rarely applicable to French movements. This is owing to the fact that the thinness of the great majority of these latter renders it unsafe to only leave a very slight endshake, as is done in England, and is necessary with such conical pivots.

The play of the pivots should amount to one-tenth of the diameter of the holes. The difference between this amount and that recommended for cylinder pivots (**415**) will doubtless be noted. This is only reasonable but it can easily be explained; the inter-dependence of the several parts or, in other words, the pitch, is so very slight that the least change in the centres of movement has its effect in altering the rate of a duplex escape-

ment. Besides this, the balance is somewhat heavy and, since the impulse is communicated to it under very satisfactory conditions, the gradual thickening of the oil has much less influence than in the case of the horizontal escapement. It is thus evident why the play of the pivots of a duplex should be reduced as much as possible.

Practical Details on the Balance-Spring.

556.—One reason for the excellent rate obtainable with this class of movement is the action of the balance-spring, an action which continues for a longer period and is less constrained than in a cylinder watch.

Moinet observes in his *Traité d'horlogerie:* "The duplex escapement admits the use of both an isochronal balance-spring and a compensated balance."

It is unquestionably true that these two correcting influences can be applied to it; but they add so little to its uniformity and so very materially increase the cost and difficulty of timing, which can then only be properly done by the skilled hand of a chronometer springer, that we are compelled to regard them as useless and even harmful, so seldom is it that they satisfy our reasonable expectations.

We have met with many duplex watches that possessed an excellent rate, even though only provided with the ordinary balance.

As a general rule, escapements in which a sort of natural compensation exists between the several frictions should not be supplied with a balance-spring of too great length. There is one definite length to be found by experiment, that is more satisfactory than any other (**349, 350**).

A spring of about ten coils is usually best; for it is to be observed that, although a very long spring may have been necessary with the extended arcs of vibration formerly in favour, it is no longer so at the present day. At the same time this length of spring must only be regarded as a rough approximation to the exact amount, for by a mere change in the position of the external point of attachment the number of coils may be varied without altering the actual length of the spring employed.

It is very important that the spring possess uniform strength throughout, that its coils be equidistant, and that the internal extremity have a gradual bend towards the collet

without any sharp angle, so that it may open and close regularly. Careful study and practice can alone convince the workman how much this curvature influences the prolonged uniformity of movement of the spring. The more a balance-spring fails to conform to the above conditions the less *life* will it have; and the rate of the timekeeper will be, in proportion, less satisfactory.

557.—Watchmakers are not agreed as to the best position to adopt for the neutral point of the spring.

The majority place it so that when the mainspring is let down the roller notch is exactly opposite the centre of the wheel. The balance is thus required to move to the same distance on either side in order to admit of the entrance and exit of the tooth.

Some prefer that the neutral point come midway between the two lifts, so that the roller is held in the position it occupies at the conclusion of the small lift.

Finally the escapement has sometimes been set so that, when at rest, a resting tooth is on the point of entering the roller notch. One would think, however, that such a practice should only be resorted to when the motive force is very great.

Each watchmaker must decide for himself as to which of these arrangements is best suited to the escapement he constructs or repairs. (See the articles on the *Balance-Spring* in the third part of this work.)

Practical Details on the Balance.

558.—The duplex escapement requires a balance that is rather larger and heavier than that employed with the cylinder, for with the former the lifting action takes place under far more favourable circumstances and is more energetic; besides which this escapement has only been found to give its best results when the motive force is somewhat great.

As we have just seen, a compensation balance is not essential to this form of escapement.

With a light balance the watch can never possess a uniform rate; when too heavy it involves the application of an excessive motive force, besides increasing the flexure and vibration of its staff, and thus rendering the risk of breakage greater. The best mean between these two extremes is the more difficult to determine according as a marked increase in the extent of the vibrations is observed to take place on diminishing the weight of a balance when it is, in the first instance, too heavy; but this

reduction must take place through a considerable range. This practical observation has led some to conclude that the balance which secures the most extended vibrations is preferable, since it reduces the risk of breakage and the wear, while at the same time the increased velocity compensates for the lightness; but the adoption of extended vibrations, so long in favour and apparently founded on the circumstance above referred to, was abandoned by the English. This was owing to the fact that experience had proved that long vibrations, especially when the balance is somewhat light, are apt to allow two teeth to pass at a time during several successive vibrations while the watch is in wear; whereas, with a heavy balance and shorter arc of vibration, this effect is not observed, or, at any rate, is at once checked.

These facts are sufficient to justify the system practised in England, a system that has so sorely puzzled the French and Swiss watchmakers, who are highly skilled in the manufacture of ornamental watches, intended to please the eye but not to be reliable timekeepers.

It may happen that the balance, if somewhat heavy, renders the escapement more difficult to time in position, and it then becomes necessary to set the balance out of equipoise; this practice has often been resorted to in England in the cases in which the pivots of the balance-staff are not conical, for the workmen know that in such cases it is of the first importance to reduce the effect produced by jerks on the vibrations; and this result can only be secured in English watches by employing a balance that is somewhat heavy and may even have to be large.

But it must not be forgotten that a balance may be too heavy in proportion to the motive force and, at the same time, too light when compared with the resistances which occur in the escapement; this latter amounts to its being too light to secure a proper rate. When such is the case the only possible remedy is to increase the *mass* of the balance and the motive force in proportion.

559.—Workmen are usually guided by an empirical rule that requires the *size* of the balance in a duplex escapement to be the same as that of the barrel cover (or slightly more). Some assert that its diameter should not exceed twice that of the escape-wheel.

Such proportions, deduced from escapements that were found to go well at first, are only useful as approximations. In

this, as in the cylinder escapement, a certain relation has to be found between the size of balance and the radius of friction, that is of the roller; when this relation is established it has the effect of rendering the escapement but little influenced by variations in the motive force and the thickening of oil, as we have already explained in discussing the horizontal escapement.

In practice, workmen ascertain whether the balance is of correct weight, so as not to be too heavy for a given motive force, by causing the balance to recoil with a piece of pegwood until the resting tooth just enters the notch, and then liberating it. The lift should suffice to give the requisite motion without undue force.

A safer method is that which we have already explained as *Half-Timing* (**440**, **441**). After a little practice the watchmaker will be able by this method to ascertain with considerable rapidity whether the weight of the balance is correct and the escapement sufficiently insensible to variations in the motive force. (See the chapter on the *Balance.*)

Number of Vibrations per hour.

560.—A duplex watch is generally made to beat 18,000 vibrations per hour.

At one time the English used to make the diameter of the impulse wheel two-thirds that of the great wheel. It had thirteen teeth and gave 14,400 vibrations in an hour. The balance moved slowly and was provided with a weaker balance-spring than that necessary for 18,000 vibrations and was very apt to trip.

One would expect that by increasing the number and therefore the rapidity of the vibrations it would be so much easier to avoid these inconveniences that we should be obliged to employ a balance-spring that is stronger, and therefore less under the control of the balance itself. Experience confirms this reasoning, and indeed duplex escapements giving as many as 21,600 vibrations have been found to be accurate timekeepers, notwithstanding that those who wore them often went on horseback. The wear of the rubbing surfaces was no more serious than with 18,000 vibrations (**39**).

Verification on the Depthing Tool.

561.—Having affixed a finger or some form of pointer to indicate the degrees of lift on the staff of the roller, the escape-

ment is placed in a depthing tool one of whose centres is provided with an appliance such as that represented at A B D, fig. 30, page 230, except that the graduation must extend to 80°.

Gently close the two branches of the tool, while guiding the roller with one finger and the wheel with another, until one of the resting teeth just rests on the roller; then deepen the pitch as much as is thought fit, but care must be taken to leave sufficient freedom between the end of the impulse pallet and the impulse tooth near which it has to pass.

The *small lift* is verified by turning the roller (by means of the index) up to the point at which the resting tooth escapes from the notch. After observing the degree indicated by the pointer, the great lift is allowed to take place, and then the roller is moved backwards until the succeeding tooth falls into the notch. The interval between this second drop and the point previously recorded should not be less than 20°.

Proceed in a similar manner to verify the first drop; that is to say at the instant the resting tooth escapes from the notch and the impulse tooth engages with the pallet, move the finger backwards until the impulse tooth escapes. It should traverse an arc of about 10° during this backward movement (**545**).

As the impulse pallet is only held by friction on the staff it can be easily set in the proper position for receiving with certainty the impact of the tooth, allowance being at the same time made for a drop of 10°.

The escapement is then again caused to act, in order to make sure that the tooth engages with the impulse pallet during an arc of not less than 20° or more than 35°, the exact amount depending on the several dimensions of the escapement; for, as we have seen, this lift (inclusive of its drop) varies in extent from 35° to 50° according to the greater or less difference between the diameters of the two wheels.

The second drop, immediately succeeding the great lift, should be no more than is necessary to ensure the proper action of the several parts.

If it is found to be too great, this is due to one of the three following causes; impulse teeth in wrong position, pallet too short, or pitch of resting teeth in the roller notch too shallow.

When the small lift is insufficient the pallet will be found to be too long; for it renders the deepening of the pitch impossible.

With an excessive small lift the pallet has been made too short either through necessity or want of care; the former is the case when the impulse teeth are out of place.

Should the great lift be too great, in consequence of the impulse wheel being larger than is required, it can only be corrected by renewing the wheel; for if we attempt to reduce the lift by making the pitch less deep, the drops will become excessive.

Finally with an insufficient great lift reduce the teeth in the manner adopted for removing the burrs (**553**), and bring the centres of movement nearer together.

Observations.

The workman may make sure that the points of the teeth do not touch the bottom of the roller notch by slightly deepening the escapement, and this will also enable him to see that the tooth has sufficient freedom to give it a minute shake at every period of its passage.

In passing by the tooth v (fig. 3, plate II.) the pallet should have enough freedom to avoid all risk of the two in any way coming in contact when the position of the watch is altered, or when the centres of motion are brought as much together as the play of the pivots in their holes will permit.

Similarly the space between the pallet and the tooth R should be verified, for, as they gradually approach one another, the interval between them becomes very minute towards the termination of the small lift.

With a view to afford additional security when these two pass near each other, some watchmakers, instead of placing the impulse teeth midway between the resting teeth, place them a little to the left. It is better, however, to keep them exactly in the middle.

After completing all these corrections and satisfactorily verifying the escapement it must be planted in the ordinary manner.

Verification of the Escapement when in Position.

562.—When the lifting points are not marked on the plate, make two dots separated by an arc of 60° and divide this into arcs of 10° each. A mark on the rim of the balance should be over the second dot when the great lift terminates.

After making sure by means of an eyeglass that the pitch of the pallet and impulse teeth is sufficiently deep, and that

those parts which should not be in contact have the requisite freedom, the escapement may be verified as explained in the preceding article. Or, if it have previously gone satisfactorily, a verification as follows may suffice.

The points of the teeth, the roller and the pivots are cleaned, the pivot-holes oiled and the escapement put together but without the balance-spring; having wound up the watch the balance is guided with a pegwood point to about the middle of the small lift and then released. It will perform a number of vibrations without stopping, during which period the motion of the wheel must be carefully and *constantly* observed. Placing the watch to the ear and varying its position, if it be found that two teeth never pass at a time, that the drops are equal, that the sounds are uniform and not interrupted by a grating noise, etc., we may conclude that the action of the escapement takes place with sufficient regularity. (Half-timing may be accomplished in this manner, but an easier method is that explained in articles **440** and **441.**)

Timing in Position.

563.—When timing in position it is only necessary to follow the instructions given in article **431**; and, for rapidly regulating, those contained in article **432** and the following. (See the chapter on *Springing and Timing.*)

Over-Banking.

564.—There is no occasion to fear over-banking with the duplex escapement; but, doubtless with a view to avoid tripping (see the causes of stoppage), it was formerly provided with a movable banking-pin. This, however, was somewhat inconvenient and has been suppressed.

CHAPTER IV.

CAUSES OF STOPPAGE AND VARIATION IN THE DUPLEX ESCAPEMENT.

Setting.

565.—Setting, if it occurs with a duplex escapement, results from a cause that is permanent and not accidental; but, although it cannot be entirely removed, it can be very materially reduced.

A balance that is too heavy, a notch that is too narrow or

whose edges are rough, a balance-spring that is carelessly fixed, and above all an excessive small lift, rendering necessary a considerable movement of the balance before the unlocking takes place, all tend to increase the intensity of setting.

When the play of the teeth in the notch is sufficient, the angles and rubbing surfaces well polished, the balance-spring so fixed that, in its neutral position, the roller notch is exactly opposite the centre of the wheel, and when the small lift is between 20° and 25° (or 30° as a maximum), the escapement will be easily set in motion and the mere shake occasioned by winding will in nearly every case suffice to move the balance.

Increasing the small lift somewhat reduces the friction during rest, and the extent of the arc of oscillation will thus become rather greater; but this is of little importance compared with the fact that, when it reaches, say, 60°, rapid and very long vibrations are essential in order to avoid setting when subjected to the mere shaking that must occur in the pocket of the wearer. And we shall presently see that long arcs of vibration are objectionable.

An Escapement that trips.

566.—One great fault to which this form of escapement is subject is that of allowing two teeth to pass at a time whenever it is shaken: this action is usually expressed by saying that the escapement *trips*. It results from a too rapid increase in the velocity and extent of the vibrations. The watch of course gains at an immense rate. If it be shaken with a circular movement, being at the same time held to the ear, it will trip and the double blow can be distinctly heard. This slipping of the teeth often continues for some seconds after the shake has occurred.

It results from the following causes: vibrations of too great extent or not sufficiently frequent, lifts too much prolonged or excessive motive force, or a balance that is too light. This last is the most frequent cause. With a heavy balance the escapement can unquestionably trip, but in that case it is only momentary.

The English succeed in avoiding this fault by making the roller larger, the balance heavier and of increased diameter, and by reducing the complete arc of vibration. They thus set a reduction in the amplitude against an increase of both size and weight. (*See* the Introduction,—article **24**, on Momentum.)

To replace a broken Balance-staff.

567.—Every watchmaker knows how to make a new balance-staff. First ascertain the several heights either by reference to the old one or, in its absence, by the method explained for the cylinder (**463**). Holding the small brass rule vertically over the lower ruby, press a resting tooth, previously covered with rouge, against it. Make a notch in the rule (M, fig. 41) to indicate the height of the roller and, previously to doing this, cut a small tongue which, when the rule is held in position, should meet the impulse tooth just as the pallet is required to do.

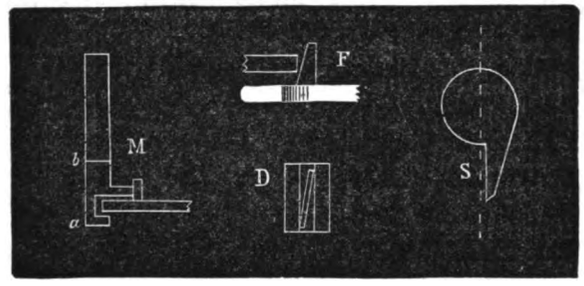

Fig. 41.

The interval between the top and bottom ruby having been measured, it is indicated on the rule by the mark *b*. It will then be only necessary to add the thickness of these two jewels to the space *a b* in order to ascertain the total length of the balance-staff with its pivots.

The best height for the balance must be ascertained in the manner already explained, and need offer no difficulty. The same may be said of the pivoting (**473**).

The staff should be thicker than the roller at the point at which the pallet is attached, so that, when this latter is removed or replaced, there may be no risk of damaging the roller.

The portion within the roller should at most be half the thickness of the roller itself. Of course when the roller is ready made the diameter of the perforation will fix the size.

If the roller is useless it is well to ascertain whether the diameter of the staff cannot be slightly increased; such an increase is usually made by some makers as we shall presently explain.

Most Swiss manufacturers make the roller axis very thick, and the roller itself, becoming exceptionally thin, is completely

divided by the notch. They assert that this does not lead to more frequent accidents because the increased thickness and rigidity of the staff fully compensates for the delicacy in the roller.

Pallet too long,—too short,—too much cut away.

568.—When the pallet is too long there is considerable risk of catching. If of steel it can be easily shortened. When jewelled and a lapidary is not accessible, the notch in which the jewel rests must be cut somewhat deeper.

A short pallet when of steel must be replaced; but if there is a jewel it will only be necessary to advance it a little.

When the pallet is cut away beyond the radial line, as shown at s, fig. 41, page 339, it engages with the tooth in an oblique direction and gives rise to considerable engaging friction.

The corner of the pallet should be very smooth and delicately rounded so as just not to scratch the nail.

There must be sufficient freedom between the flat of the wheel, the resting teeth and the under face of the pallet. This latter is always gently driven on to the balance-staff; it must on no account be loose.

It is a good plan to make two notches on opposite sides of the collet that carries the pallet in order to be able to grip this collet in a pair of cutting pliers (the edges of which are slightly rounded) when the pallet requires to be rotated on its axis. When pliers are used for this operation there is danger of scratching the corner of the ruby or even breaking the balance-staff should they slip off the pallet.

A cracked, broken or loose roller.

569.—It is no uncommon occurrence to meet with rollers that are very slightly cracked, which, although not causing a stoppage, give rise to constant irregularity.

The roller should be made of ruby. If it be absolutely impossible to replace a broken roller by one of stone, it must be re-made in steel hardened in opium or mercury and well polished. But, as Moinet has observed, rubies are always the safest.

When renewing a roller it is well to make it first in brass, both in order to try it and as a pattern for the lapidary.

570.—In fixing anything in position by means of shellac, whether it be a roller or an impulse pallet or indeed any part of

an escapement, such as the body of a ruby cylinder, the pallets of an anchor, etc., instead of rubbing the heated surfaces with a piece of shellac, it is better to previously dissolve some shellac in spirits of wine (alcohol) and, using a small paint-brush, to spread it over the surfaces that are intended to adhere. It has been observed that parts joined in this manner are held more firmly.

Notch too large,—edges too much rounded,—narrow notch.

571.—When the notch is too large or the edges too much rounded the recoil will be excessive, the lift insufficient, and there will be some danger of teeth slipping when they are not of full length. Such faults render an increase in the small lift necessary.

If the entrance edge is too much rounded it gives rise to an augmentation of recoil. When a similar fault occurs at the exit edge the point of the tooth is liable to occasionally fall on to this rounded edge, and is then driven, under engaging friction, back into the roller notch; the effect is very detrimental.

With a narrow notch any particle of dirt falling into it, or even the mere thickening of oil, will alter the rate of the watch and may cause it to stop.

A wheel unequally divided.

572.—The resting-point must be constantly changing when the wheel is out of truth or unequally divided; and the same may be said of the energy of drops and impulses and of the extent of the lifts. With the shorter teeth, the small lift will be uncertain and the friction of the resting tooth will be excessive, since it occurs near the line of centres. With the longer teeth the small lift will be of too great extent and the risk of setting will therefore be increased.

If the impulse teeth are not equidistant there will be a reduction in the great lift; for we should then be compelled to employ a shorter pallet than when the wheel is accurately divided. Hence it is important to verify each tooth of the wheel.

The remedy for all such faults is to replace the wheel; but if it is absolutely essential to retain it, the resting teeth must be trimmed at their points (**553**) in order to make the wheel round, and so far to diminish its diameter as compared with that of the impulse wheel that, when the escapement is planted afresh, the correct performance of the great lift may be guaranteed and its extent may be sufficient (**537**).

Impulse teeth wrongly placed.

573.—When the impulse tooth is not midway between two great teeth, the pallet will pass too near one impulse tooth and too far from the other; it therefore necessarily follows that the pallet must be made shorter than the prescribed amount, and it is often found impossible to obtain a great lift of sufficient extent.

Formerly, when there were two distinct escape-wheels on the same axis, it was a simple matter to rotate one or the other on this axis and so adjust the position of the teeth; but at the present day, with a single wheel, if the impulse teeth are improperly placed, we can only bend the long teeth in such a manner as to bring the impulse teeth midway between their points. This delicate and even dangerous operation may be performed as follows:

At the centre of a circular plate (fig. 13, plate V.) about $1\frac{1}{2}$ inches in diameter, a projecting boss or shoulder is left on which the wheel can be firmly fixed by screws as on a chuck. The resting teeth project completely beyond the edge of this boss. The surface of the plate is divided radially into some multiple of the number of teeth of the wheel, say fifteen or thirty parts, and it is then easy, by means of a perforated punch that is pressed on to each tooth, to bend them all to the same extent, since the divisions on the plate serve as a guide in inclining the punch as well as to ascertain whether the points are sufficiently bent.

The wheel is then trimmed and trued with very great care (**553**).

Small Lift too great or too small.

574.—An insufficient small lift, besides rendering the action of the escapement insecure, causes the rest to take place too near the line of centres (**74**) and the engaging friction is excessive.

If the escapement acts with certainty in every other particular and in accordance with the principles above laid down, it will be enough to replace the roller by one slightly larger, instead of making the pitch of the teeth deeper; and the converse will be the case with an excessive small lift (providing the wheel be not trimmed); for when the small lift is too great, setting is very liable to occur, and a single shake will sometimes suffice to occasion a stoppage.

Swiss escapement-makers correct any faults resulting from a slight error in the planting of the escapement by altering the roller; with them this is an operation of no difficulty, since they are always provided with ready-made rollers, sized according to a recognized scale.

Great Lift excessive or insufficient.

575.—We have already shown that any deficiency in the great lift results either from the impulse wheel being too small, or the pitch too shallow, or the impulse pallet too short.

We have also seen that an excessive great lift is occasioned by the impulse wheel being larger than is required, or the centres of movement too close together, this latter cause, moreover, gives rise to an exaggerated small lift.

If the great lift is deficient, the watch cannot be regulated.

If excessive, there is risk of stoppage and tripping: the former when the motive force is feeble; the latter when it is great.

In the first case the depths must be verified and the mainspring changed; on the other hand, in the second case the motive force should be reduced, or rather the weight of the balance increased, for if the pallet be shortened the drops will become greater, and if the pitch be made shallower the pallet will be too short and must be replaced.

It is impossible to do more here than draw attention to the faults and leave it to the intelligence of the workman to discover and apply the best available correction; we would merely remind him that, with very few exceptions, a contact between the pallet and impulse tooth of not less than 20° (exclusive of the drop) best satisfies the requirements of practice.

The Fourth Wheel Depth Bad.

576.—The escape-wheel pinion depth should receive the greatest possible care: it has a very marked influence on the timing. Unfortunately this pinion nearly always has six leaves, and those who plant the escapements in factories are for the most part utterly ignorant on the subject of depthing: the result is that this depth is generally the worst in the whole watch.

An equal amount of care should be devoted to all the other depths, for we have met with cases in which a duplex watch

could not be regulated merely because one depth in the train was rather too shallow; and yet this depth felt quite smooth and did not occasion stoppage.

Other Causes of Stoppage and of Irregularities in Timing.

577.—The other principal causes of stoppage and variation in the duplex escapement are:

A WHEEL BADLY PLANTED. It will be liable to rub against the passage of the escape-wheel cock or against the last wheel of the train. If inclined to one side the resting teeth will not be square with the edges of the roller-notch, will act as though they are too thick and will be apt to catch. This latter fault may also occur if the faces of the teeth are not cut square, that is to say in a vertical plane to that of the wheel (D, fig. 41, page 339).

TEETH TOO THICK OR NOT OF UNIFORM THICKNESS, etc.

IMPULSE TEETH WHOSE FACES ARE NOT CUT SQUARE. The freedom between the pallet and tooth may vary with changes in the position of the watch and especially if the endshake of either mobile is at all in excess (F, fig. 41, page 339).

POINTS OF THE TEETH TOUCHING the plug on their lower side or the pallet on the upper.

IMPULSE TEETH WORN ON THEIR RUBBING FACES AND A PALLET THAT IS PITTED OR HAS LOST ITS POLISH WHERE STRUCK. Such a case is by no means rare with a steel pallet and is also met with, though less frequently, with a ruby pallet.

A ROLLER OUT OF TRUTH, arising from its being either carelessly made or badly cemented. The point at which rest occurs is continually changing and the wheel is alternately caused to advance and recoil. Thus at one period of its action the escapement will exhibit those faults that result from a too small roller and at another those due to one that is too large.

If the performance of the small lift is not satisfactory through the size of the roller being insufficient, this roller may be gently heated and the side on which the notch is cut then pressed as much outward as possible, or the converse may be done when the small lift is too great; but such a practice must only be resorted to as a makeshift.

A ROLLER THAT IS BADLY POLISHED OR PITTED by the teeth (when of steel) or ROUGH especially at the edges. It must always be carefully examined, after cleaning, under a powerful eyeglass.

A ROLLER OR IMPULSE PALLET BADLY CEMENTED or loose.

A ROLLER CRACKED transversely or longitudinally.

JEWELS LOOSE IN THEIR SETTINGS.

A LOOSE PLUG (**570**).

PARTICLES OF SHELLAC either in the roller-notch or on the points of the teeth.

PIVOT HOLES TOO LARGE—OR TOO SMALL.

EXCESSIVE ENDSHAKE of either the balance or escape-wheel, especially with conical English pivots (**555**).

CURB-PINS THAT MOVE WHEN TOUCHED BY THE BALANCE-SPRING.

Concluding Observations.

578.—As a considerable number of causes of stoppage and irregularity have been pointed out in the three preceding chapters it is unnecessary to again refer to them. Others again are common to all forms of escapement, so that the above list may be rendered more complete by selecting, from Chapter VI. of our treatise on the Cylinder Escapement, such cases as are applicable to the duplex: we will conclude our consideration of this latter by the following observations.

In the humid climate of England the performance of the duplex is more satisfactory than that of any other escapement. When the pivots are conical and extremely fine, so as to require very little oil, the duplex will continue in action much longer than either the cylinder or lever.

In carriage clocks it has given better results than are obtainable with most other escapements; but the duplex has in nearly all these cases been manufactured in either London or Paris. As to those of Swiss construction they have been very generally given up by makers of such clocks, as they failed to give complete satisfaction.

These adverse results are apparently due to two principal causes: the arcs of vibration are too much extended and the diameters of the roller and balance are not correctly proportioned.

NOTES

ON VARIOUS UNDETACHED ESCAPEMENTS.

579.—Before leaving the consideration of escapements of this class we must refer to a few forms occasionally met with.

We will, however, confine ourselves to a brief description, followed by a few quotations and general observations.

VIRGULE OR HOOK ESCAPEMENT.

580.—Two forms of escapement so named have been employed: one, the *double hook*, is extremely difficult of construction and, as at the present day it is no more than a rare

Fig. 42.

curiosity, we shall not consider it; the other known as the *hook* or *virgule* escapement will now be briefly discussed.

Action of the Escapement.

581.—A flat wheel is provided with long sloping teeth (*i* and *d*, fig. 42) whose extremities are turned upwards so as to form small prismatic or semi-cylindrical pins. These are represented in horizontal section at *i* and *d*.

The principal piece, the virgule or hook, is shown in plan at *h f*, and derives its first-mentioned name from its resemblance to the comma (French, *virgule*), the sign of punctuation. It is fixed to the balance-staff; indeed both are formed out of the same piece of metal.

Above the hook this axis is necessarily made weak since a deep notch has to be cut to allow of the passage of the teeth. The shaded portion v indicates the part of this staff that is not cut away, which is known as the *crank*.

The *outside rest* of the tooth takes place against the semi-circumference *h* and this must be somewhat projecting in order to avoid contact of the entire surface of the pin with the balance-staff.

It commences when the hook, travelling from *f* towards *j*, has moved to a sufficient distance beyond the latter point to allow of the escape of a tooth from *c*. It terminates during the return motion from *j* to *f*. For during that period the tooth is pressing on the entrance edge *b* and forces it backwards, producing the *small lift*.

This first lift is followed by the drop of the tooth against the inside of the small cylindrical cavity *c*, when the *second* or *internal rest* takes place.

The balance is now brought back by the balance-spring and returns until, when in the position v *j*, the second rest terminates, and the tooth *c*, being free to move, presses on the surface of the long arm *i f* (then in the position *j*), and this action constitutes the *great lift*. The succeeding tooth now goes through the same series of actions.

582.—The form of the arm *i f* was such as to render the lifting action uniform. It was usually a mere arc of a circle of the same radius as the escape-wheel. This curve was struck from a centre rather below that of the wheel and such that, when the great lift was just commencing, the angle between the external circumference of the wheel and the curve of the hook (in about the position v *j*) was 30°. A modern author, who has asserted that a curve of such a radius will not secure any lift, is mistaken.

Two methods were formerly in vogue for making the hook: by one the staff, pivots and hook were all formed out of one piece of steel, the internal resting surface being made with a circular cutter; in the other and more practicable method, the

hook is formed of a piece of steel perforated to a distance equal to the interval between the lower pivot and the crank v with a hole slightly larger than one of the pins; a separate axis is made and on this the lower pivot is turned, or, in view of any necessity for removing this pivot, a plug may be made analogous to those in a cylinder.

The total length of the hook is equal to the interval between two teeth, an allowance being made for freedom.

The entrance edge *b* should be such as will give a movement of 10° to the balance, and the great lift should measure 30°.

When the balance-spring occupies its neutral position, the tooth should be commencing either the small or great lift: the escapement is not liable to set. The most prominent point of the external face of the pins should pass through the centre of the balance-staff.

Summary of Authorities.

583.—JURGENSEN.—This inconvenience (the friction that occurs against cylinders) suggested the idea of the hook escapement, in which the friction would become unimportant if only the oil could be retained on the rubbing surfaces; but experience has clearly proved that the oil only remains where required for a short period and is then generally attracted elsewhere; hence this form of escapement cannot be employed with success notwithstanding the *excellence of the principles* on which it works (**77**).

584.—TAVAN (1831).—The two lifts are of very different kinds. In the first, on entering, the pressure is borne on the inclined face of a lip (*b*, fig. 42, page 346) that is comparable to the entrance lip of a cylinder, the thickness of which is considerably increased. The tooth meets this surface when the two are travelling in directions opposite to each other, and this action is detrimental both on account of the nature of the friction and of the form of lever, for the effective force is diminished on approaching the centre at the same time that the resistance opposed by the balance-spring increases (**304**).

The two lifts are applied through the interposition of levers of very different lengths; the two rests also occur at very varied distances from the centre of movement of the balance. Perhaps this failing appears more important in theory than it actually is in practice, for watches provided

with this form of escapement have certainly been proved to possess excellent rates.

It appears to have two advantages over the cylinder; one is the great energy of the second lift which renders it possible to employ a heavier balance and a corresponding balance-spring, or, in other words, to exercise a more effective control over the motive force. The second advantage lies in the fact that the friction during rest is less than in the cylinder and less influenced by the thickening of oil. But in view of the difficulty of construction of this form of escapement and the facility with which it gets out of order, it has for a long period fallen into disuse.

585.—MOINET (1847).—"Although employed with success by some of the best makers and notably by Lépine, whom some consider to have invented it (while others give the credit to Lepaute), the virgule escapement has been almost universally abandoned in favour of the cylinder."

Moinet quotes the following passage from Jurgensen: "Its main fault consists in the difficulty of retaining oil on the rubbing surfaces; and when this is wanting they soon become worn and the rate ceases to be uniform;" and he adds, "care, however, may remedy this defect for we have seen hooks made by Lépine remaining as good as new after being in use for a very long period, and the present author believes that a steel wheel might keep the oil more pure: the wheel is usually of brass. If the teeth were somewhat high and the oil carefully placed, its spread might be prevented." (All these remarks had been already made by Jurgensen.)

OBSERVATIONS.

586.—When the hook escapement is thoroughly understood and well made, in accordance with the laws that govern escapements in which the rest is frictional, the watch can be timed quite as well as an ordinary cylinder; but such a result, no more than an equivalent, can only be secured by the application of much more care and delicacy, and the first of these escapements has therefore with reason been replaced by the second.

Two principal causes have, then, led to the disuse into which the virgule has fallen.

(1) Its wear was very rapid;

(2) The rate became irregular after the lapse of some time.

The excessive wear arose from the fact that in the great majority of cases the hook did not retain the oil, and this,

being only present on the pins in very small quantity, dried up rapidly (**92**).

The want of uniformity resulted from the increase in friction upsetting the proportion that previously existed between the several parts of the mechanism (**364**), and from the fact that the impulse pallets were constructed in accordance with that erroneous theory which requires their form to be such as shall procure a uniform lift; whereas this form, in nearly all cases, increases considerably the energy expended in the drop (**272**), and thus renders the virgule analogous to a cylinder escape-wheel with inclines of excessive curvature.

These faults might have been avoided (1) by forming each lifting surface so that it retained the oil and did not permit a too rapid movement of the wheel; and (2) by arranging the pins in the manner adopted by a clever watchmaker, Henry Savoye, nephew to A. Breguet.

Over each tooth in succession he placed a hollow cutter, similar to a riveting punch, finely roughed on the inside like a file to the requisite depth. A double movement was then given to it; while being rotated on its own axis a kind of rocking motion was set up analogous to that adopted when finishing off the ends of holes in jewels. By this means the pin was formed as shown in elevation at *n* (fig. 42, page 346). The point of contact of the hook was in this case at *n*.

Providing the oil is applied with proper care, it is retained very well at a point of contact so formed (**91**). Several escapements made in this manner by Savoye have gone for more than twenty years without showing any appreciable signs of wear.

We will conclude with a brief summary of the contents of this chapter.

The hook escapement when constructed by a skilful and intelligent watchmaker will give results that are sufficiently satisfactory for ordinary use. So long as it was made at Paris or by Lépine at Ferney, it was well thought of; but when taken up by the manufacturers it soon fell into disrepute.

It cannot be considered as equal to the cylinder. Even when only moderately well made this latter may possess quite as good a rate, and it has over the virgule several advantages; it is less fragile, more easily made, requires less care in construction so that it can be produced in factories at very low cost, and finally it may be plentifully supplied with oil.

BREGUET'S RUBY CYLINDER ESCAPEMENT.

587.—This form of escapement, first made about sixty years ago, differs from the ordinary cylinder both in the shape given to the cylinder itself and in that of the wheel.

The complete cylinder consists of three parts: the axis *b a* (fig. 43), the steel setting H *c d*, and the ruby body T.

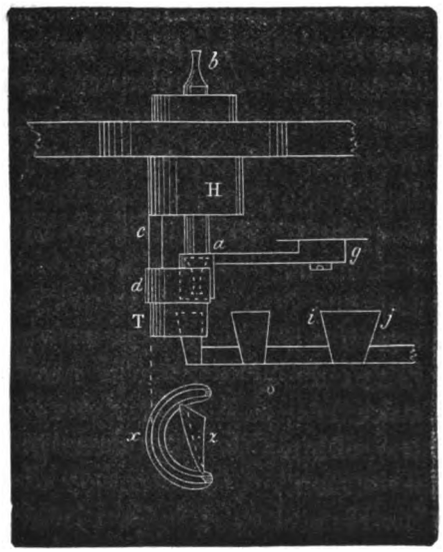

Fig. 43.

The setting is attached to the axis by means of a cylindrical ring or collet H; the pillar *c* connects this collet with the half-cylinder *d*, in which is cemented the ruby body T.

A horizontal section of this half-cylinder with the body in position is shown at *x*.

The balance-staff terminates in two pivots slightly enlarged at their extremities in order that they may the better retain oil, and to reduce friction when in a horizontal position. The upper pivot works in the usual balance cock while the lower one is received in a jewelled hole in the part *a* of the small bar *a g*. When the vibrations are of too great extent the pillar *c* will rap against this bar.

On this system it will be seen that the body is external to the balance and its staff, that is to say if the escapement is inverted the body is entirely exposed.

The teeth are not inclined planes protruding beyond the flat

of the wheel; it is a simple crown wheel whose section may be regarded as a ring cut from a regular cone.

The form of teeth will be at once understood from an examination of fig. 43, where they are shown in elevation at T and *j* and in plan at *z*. The extreme width *i j* is the same length as the inclined plane in the ordinary escapement.

During the rest the rounded corner *j* of the tooth presses against the internal or external surface of the body, and an impulse is given by the top *i j* of each tooth passing in contact with one or the other lip; this top is a slightly curved incline or rounded in a beaded form; it acts in precisely the same manner as the impulse plane of an ordinary tooth.

Remarks.

588.—This escapement was abandoned because, while possessing important qualities which might be useful in other escapements, it had several disadvantages.

The good qualities lie in the form given to the wheel; it can be very easily made and its shape is so well adapted for retaining oil on the rubbing surfaces that it always returns to them whenever forced away by the action of the escapement, even until the oil is completely dried up (attraction exerted by angles—**85**).

The faults consist in the great care necessary in making the cylinder, and more especially the lower pivot hole of the balance; the smallness of the play that can be allowed to its pivots, and lastly the difficulty experienced in retaining oil in the lower pivot hole. If it comes in contact with the corner before reaching the bottom it spreads and dries very rapidly.

The inventor succeeded in overcoming this difficulty by employing an instrument similar to the drilling tool. The hole was set centrally on a plate by means of one end of a delicate centre, and fixed in position; this centre was then reversed, and the other extremity, formed like a drill, was employed to deposit the requisite amount of oil.

The escapements with frictional rest employed in clocks will be considered separately in a subsequent portion of the work.

NEW THEORY OF

DETACHED ESCAPEMENTS.

THEIR ORIGIN.

589.—The detached escapement is dead-beat, but its principal characteristic, distinguishing it from escapements with frictional rest, consists, as we have already seen, in the fact that the vibration of the balance takes place in complete independence of the escape-wheel except during the very brief period of lift. This wheel does not rest against any part of the axis of the balance but against a separate intermediate piece. The lever escapement in watches, the detent or chronometer escapement, and several forms of escapement used in timepieces are of this class.

The invention of the earliest form of detached escapement is thus referred to in the *Mémoires de l'Académie des Sciences* for 1748:

"A novel form of dead-beat escapement, invented by M. Le Roy, junr. (Pierre Le Roy). Whereas in the dead-beat escapements hitherto employed the wheel presses against some piece rigidly fixed to the balance throughout the entire oscillation, and friction therefore must take place between that piece and a tooth, in this new arrangement the wheel falls against an appendage (the detent) which is fixed to the watch-plate entirely disconnected from the balance, and is held there at each half-revolution."

The first useful application of the detached escapement to the measuremeut of time and the determination of longitude at sea is also due to Pierre Le Roy. This was on the occasion of his completing a marine chronometer in 1766.

We are careful to give these dates accurately because F. Berthoud, in the eulogy of his own discoveries which he entitled *Histoire de la mesure du temps*, asserts that the detached escapement was first experimented upon in 1754.

These experiments, which we need hardly observe were his own, were anything but successful. Hence he did not employ escapements acting on this principle in his two chronometers Nos. 6 & 8, the first he made that gave the longitude with any accuracy. They were not completed till 1768, and were provided with a special form of cylinder escapement in which the cylinder was connected with the balance by an intermediate rack.

It was only about the year 1770 that F. Berthoud was successful in constructing a detached escapement; and that was but a modification of Le Roy's design.

With the modesty that characterizes great inventors P. Le Roy himself tells us, in his *Etrennes Chronométrique*, that, when showing his chronometer to the sons of Dutertre, a clever Paris watchmaker, after the death of the latter, they showed him a model of a detached escapement made by their father, which, while the object in view was similar to his own, differed considerably in the details of construction.

This fact, however, cannot weaken Le Roy's claim to the discovery; for Berthoud, notwithstanding that he was likely to have it brought under his notice, declares his ignorance of Dutertre's arrangement, and was thus unable to speak of it.

The question is then conclusively settled, and it is now generally admitted that the construction of the first detached escapement and its first practical application are alike due to P. Le Roy.

PRELIMINARY OBSERVATIONS.

No Theory of Detached Escapements has yet been published.

590.—As was the case with escapements in which the rest is frictional, a theory incorporating the principles that govern detached escapements has not yet been propounded.

We certainly do possess numerous experimental data on this subject that are of real value, and some authors have gone so far as to deduce from the proportions they have found most satisfactory in practice certain requirements which they have proceeded to lay down as laws. But, for want of a fundamental theoretical basis, of a principle uniting them into one harmonious whole, these experimental data and empirical rules are perpetually in apparent contradiction, and even at the present day give rise to useless controversies.

The reason why mechanical arrangements that apparently are so essentially different give satisfactory results is never clearly explained.

The present work is designed to satisfy this want.

The following theory, like that of escapements in which the rest is frictional, does not rely on any hypothesis, but is a logical application of the laws of motion, corroborated by well-established experimental results. In not a single feature does it run counter

to the actual observations that have been published by our most skilful authors and experimentalists. On the contrary it confirms and explains them.

The same laws govern the Detached Escapements of clocks and watches.

591.—This is a precisely analogous case to that discussed in article **215**, and we need, therefore, only refer to it.

Detached Escapements are not equally independent of the motive force.

592.—We consider, in short, that such escapements differ in the relation they bear to the motive force according to their precise mechanical arrangement; and this relation may be more or less direct.

Thus the lever escapement, in which a mobile is interposed between the wheel and balance, cannot be free in the same sense as the chronometer escapement where the motive force acts directly on the balance; but either will cease to possess the principal characteristic of detached escapements as soon as any contact occurs between the escape-wheel or guard-pin and the edge of the roller. Let it be admitted then, *a priori*, that the steadiness of the mobiles when brought to a certain position by the action of the mechanism is perfect, and that no contacts calculated to invalidate the theoretical conclusions can occur.

We shall subsequently see that it is easy to realise this condition in practice, but it seems, nevertheless, well to make the observation at the outset.

Functions of a Detached Escapement.

Its balance does not act in the same manner as that of an undetached escapement.— Action of the Balance-Spring.

593.—On page 115, when discussing the escapement with frictional rest, we stated that:

"The functions which every escapement is required to perform are:

"1. To moderate and, at the same time, regulate the velocity of rotation of the train of mechanism.

"2. To restore to the moderator the small amount of force that it has lost at the conclusion of each complete oscillation.

"3. To effect this restitution of force in such a manner that all the oscillations occupy exactly the same period of time. The attainment of such a result does not in any way preclude the

possibility of employing unequal movements as regards the actual space traversed."

594.—The above may be said with equal truth of detached escapements, but a distinction must be made as regards paragraph 3.

Assuming equal periods of oscillation in the two classes of escapements, this hypothesis only asserts that the movements of the two balances will be identical.

In the frictional rest, escapement, the velocity of the balance when it is commencing the return movement, and is on the point of unlocking, is the resultant of two principal opposing forces:—the elastic reaction of the balance-spring and the pressure on the resting surface.

In the detached escapement the force that controls the velocity of the balance at different points in its path may be considered to have its origin in one simple cause, the elastic reaction of the balance-spring; a reaction which takes place with perfect freedom if the spring is truly central and isochronous, as is the case in chronometers.

The following effect results from this fact, and we shall find it of importance in what follows: the velocity of the balance at the several points of its path is represented by different figures according as it forms part of a detached escapement or one in which the rest is frictional, all other conditions being the same in the two cases.

The action of a Detached Escapement is complex.

595.—Just as was pointed out when discussing the frictional rest escapement, it may be here observed that the energy of movement of the moderator is merely the difference between two opposing forces:

The impulse given to the lifting arm;

The resistance opposed to the moving balance by the mobile against which the wheel rests.

There are then, as before, an action (the impulse or lift) and a reaction (the unlocking), but with the following points of resemblance and difference:

The impulse on the lifting arm acts in the same manner in the two classes of escapement;

The resistance opposed during the unlocking, while proportional to the radius of rest in escapements with frictional

rest, is, in the majority of detached escapements, a function of the inclination of the plane on which this rest occurs and of the length of arm carrying that plane (ignoring the friction, etc.). It thus follows that we can vary the amount of this resistance while maintaining the length of the radius of rest constant, whereas such a change could not be effected in the case of frictional rest escapements or even in such detached escapements as have the resting surfaces concentric.

PROPOSITIONS SUMMARIZING THE THEORY OF DETACHED ESCAPEMENTS.

596.—Retaining our former arrangement, we will at the outset lay down in the form of propositions the general outline of the new theory. Their demonstration will follow, and several of the experiments described will be seen to constitute an extension of the theory of escapements with frictional rest.

The reader should be thoroughly acquainted with this latter, as the two have several points in common. Indeed it could not well be otherwise since the same basis underlies both; namely the laws which govern the movement of bodies.

We might have enunciated one *General Theory*, but it appeared that the study of the subject would be rendered easier by dividing the work in such a manner as to give the different theorems in their special connection.

This being the case it will be manifest that several of the propositions given at page 116 are equally applicable here and must be reproduced. Although this repetition, which helps us to grasp the entire theory, has no inconvenience, we have nevertheless thought it well to indicate with an asterisk each proposition that is quoted literally from the former set.

FIRST PROPOSITION.

597.—If the motive force be exactly counterbalanced with an escapement arm, the conditions of equilibrium will be maintained however the length of this arm be varied, providing the lifting angle and motive force remain unchanged.*

SECOND PROPOSITION.

598.—If the lifting angle and the motive force remain constant, the force of the impulse that maintains the movement of the moderator (and regulator) will increase with any diminution in the length of the escapement arms.*

THIRD PROPOSITION.

599.—When the resting surface is concentric with its axis of rotation, the resistance offered by the unlocking is proportional to the radius of rest. The amount of energy lost by the moderator increases with the extent of rubbing surface.

FOURTH PROPOSITION.

600.—If the radius of rest remain the same, the resistance of the unlocking depends on five principal elements: the inclination of the locking plane, the virtual length of the lever by which this plane is set in motion, the direction of pressure, the extent of rubbing surface and the time occupied.

FIFTH PROPOSITION.

601.—Various combinations of these several elements may lead to identical or equivalent results; that is to say, they may either maintain the resistance to unlocking invariable or they may always retain the same degree of stability of the resting piece.

SIXTH PROPOSITION.

602.—The lift and the motive force remaining the same, the period of an oscillation will change with any variation in the length of the escapement arms.*

SEVENTH PROPOSITION.

603.—There is one, and only one, length of the escapement arms that is adapted for ensuring the nearest possible approximation to isochronism in the oscillations.*

EIGHTH PROPOSITION.

604.—The useful effect of an impulse, measured by the amplitude of the arc of vibration described by the moderator, varies with the height of the incline by which this impulse is communicated.*

We here assume the length of this plane to remain constant, because many proportions between the *height* and *length* might occasion an identical amount of motion of the moderator.

NINTH PROPOSITION.

605.—There is one degree of inclination of this incline which secures, with a given motive force, the greatest extent of oscillation with the greatest regularity.*

TENTH PROPOSITION.

606.—When the motive force is given, and the best proportion for the energy of impulse to bear to the resistance at unlocking has been ascertained for a pendulum or annular balance of known size, this proportion will require to be varied with any change in the dimensions of the moderator.

ELEVENTH PROPOSITION.

607.—The ratio between the force of the impulse and the resistance at unlocking (or rather the force necessary to unlock) varies with time.

This variation depends on the combination of three terms: the motive force, the extent of the arc of vibration, and the degree of isochronism of the balance-spring: terms which change in different proportions.

Consequently, by replacing a given motive force by one that is either less or greater, we cause both the initial ratio and its progressive change to vary.

TWELFTH PROPOSITION.

608.—The size of the escape-wheel is not a matter of indifference. It is directly dependent on the weight and velocity of the wheel, on the effective height of the impulse plane, and the friction that occurs on its surface.*

The velocity of the wheel must be made secondary to those conditions which determine the lift and the several pressures under which it occurs.

Remark.

609.—We would repeat the observation already made in paragraph **227**, to the effect that a full demonstration of these propositions is impossible except by the aid of mathematics of an advanced description; but such complete proof is not absolutely essential for our purpose. Analytical solutions possess a degree

of accuracy far beyond that which it is possible to attain in practice: and besides this we are ignorant of the exact value of several of the principal elements involved in the calculation, friction for example, and this would render any purely theoretical results open to question; we shall therefore, as in the former case, only resort to the elementary mechanical principles. Supported by experimental evidence, the solutions they afford will abundantly suffice to satisfy all the requirements of practical Horology.

But the reader must be thoroughly at home with these principles, the elements of applied mechanics, to which we have devoted our first chapter; and this preliminary knowledge should also be supplemented by a careful study of articles **228** and **229**.

THEORETICAL & EXPERIMENTAL PROOF OF THE ABOVE PROPOSITIONS.

If a given motive force be maintained in equilibrium by the arm of an escapement, it will remain so however the length of this arm be varied.

610.—Assume the lifting angle to remain constant.

The case will be precisely the same in a detached as in a frictional rest escapement, and the demonstration identical with that given in articles **230-4**, to which therefore we refer the reader. He must, however, take into account the details bearing on the subject which are contained in the following article and are applicable to the two classes of escapements.

The impelling force required to maintain the movement of the moderator increases as the arms of the escapement are shortened.

611.—As in the last paragraph, we will here assume that the lifting angle and the motive force remain invariable, and that by the expression "arms of the escapement" is understood the impulse and the locking arms.

The truth of this proposition has been proved in **237** and the succeeding paragraphs and it is equally applicable to detached escapements; it remains however for us to add certain details and to describe some additional experiments.

612.—An examination of the lever, pin, and other detached escapements shows that their modes of action satisfy the above conditions; but at first sight it appears that such is not the case

with the chronometer escapement with only one impulse pallet, on which the wheel acts at every second vibration.

The principle is, however, no less applicable in this case, and it is just as exact.

If the lift and the force producing it remain the same, it is impossible to vary the size of the wheel without modifying the impulse roller in proportion; but it is possible to alter the point at which rest occurs, as well as that at which the unlocking pallet comes into action.

It is evident, so much so that with our present knowledge its proof would be waste of time, that, with a given motive force, the moderator will lose a proportionately less amount of its energy according as the unlocking pallet is shortened, or, what amounts to the same, as the lever opposing it becomes longer.

This question requires to be considered more fully, but it would involve a discussion of the lever by which the locking pin is moved; this question cannot be adequately entered upon without some detail, and, as such details will be given in their proper place when we consider the most convenient length for the detent and the lever, we will here make only a few general observations.

613.—This principle is equally applicable to any escapement, however its dimensions be increased (providing all its parts are varied in the same proportion). With larger and therefore heavier mobiles the pressure on pivots and the extent of rubbing surfaces would become greater; hence the amount of force absorbed would be more considerable and the available energy less.

The two first propositions are in no way contradictory.

614.—It has been observed by some that the two first propositions are opposed to each other. If, they say, the energy is greater with a short arm, the equilibrium spoken of in the first proposition cannot exist.

The fallacy of such a statement can be easily made manifest.

The difference in the amount of force available for maintaining the movement of the moderator with a short or long arm is very slight, in fact it is actually less than the resistance offered to separation by two stationary bodies when pressing against each other. This action arises from: (1) their inertia; (2) adhesion between the touching surfaces; (3) the abnormal increase of friction at the commencement of motion. This

excess of resistance neutralizes the slight difference in the force and produces a statical equilibrium.

Such, however, is no longer the case if the machine is in a dynamical condition or state of motion, as then the opposing influences here referred to are absent; for (1) inertia, which in the former case tended to maintain the body at rest, now helps to keep it in motion; (2) the increased friction and adhesion on commencing motion are of course absent.

We here see another example of the errors to which the consideration of a machine solely from a statical and geometrical point of view lays us open.

EXPERIMENTS.

Preliminary Observations.

615.—Another objection has also been urged:

"Friction is proportional to pressure (**38**); but you prove that the greater pressure occurs on the incline of a short arm; hence there must result a somewhat increased amount of friction on this arm and the available energy must therefore remain the same."

This objection has no better foundation than the preceding.

When a force is expended in pressure on the surface of a body that cannot be moved in the direction of the force, the friction produced is proportional to the weight that would balance this pressure; but such is not the case when the body moves under the influence of the pressure. This being so, a greater acceleration of movement must correspond to an increased pressure.

The theoretical proof of this truth is beyond the range of elementary Mechanics; we will then only consider certain experimental evidence.

FIRST EXPERIMENT.

616.—We have constructed the apparatus represented in figure 1, plate VI. Its arrangement is sufficiently evident from the drawing and a few details will make its mode of action clear.

An escapement arm *v b d i* carries three pallets or inclines *b c*, *d f*, *i j*, so placed that their distances from the point of suspension *a* are as 1 : 2 : 3. These three inclines, having equal heights, will produce the same amount of lift.

The arm *v b d* is freely suspended at *a*; to it is attached a vertical rod *v g* carrying an adjustable sliding weight *u*.

The object in view in adopting such an arrangement was to secure great sensitiveness to variations in the pressure exerted on the pallets; for this sensitiveness, assuming it to be necessary, would be wanting if the arm *v b d* terminated at *a*, as it would then be no more than a mere pendulum. With the ordinary pendulum considerable differences in the motive force do not sensibly influence the amplitude of the oscillation when this is of moderate extent. The sliding weight placed above the centre of suspension has the further effect of prolonging the period of oscillation; and this facilitates the observations.

By means of an adjustable weight carried on the arm *t*, the arm *v b d* can be brought to rest in a vertical position.

The arm *n*, movable on pivots supported between the bearings *q*, which latter can be set in any vertical position, corresponds to the tooth of the escape-wheel; it is set successively in the positions *n n'*, *m m'*, *s s'*. These positions must be fixed upon with considerable care as the tooth is required to act tangentially at the middle of the pallet (in other words, the arm *n n'* must be horizontal).

After having adjusted the apparatus so that the line *n m s* was vertical, the first experiment was performed as follows:

The tooth *n* is placed against the middle of the pallet *b c*, and, by means of the counterpoise *r* attached to the arm *g*, the whole is brought to a state of equilibrium.

This equilibrium was found to be maintained with the tooth acting on either the arm *a d* or *a i* in the position *m* or *s*, as well as on *a b* in the initial position *n*.

SECOND EXPERIMENT.

617.—After the counterpoise *r* (fig. 1, plate VI.) had been detached, the arm *v b d* was balanced so that the points *b*, *d*, *i*, were a little in advance of the line *n s*, and it was held in this position by the thread *l* attached to the hook *k*. The tooth *n* was then, in three successive observations, caused to act on the uppermost points of the pallets, and, on burning the thread, an oscillating movement was imparted to the arm *v b d* by the passage of the tooth along the inclined plane. The extent of oscillation, as indicated by the index *g* moving over a scale, was always somewhat greater when the shortest escapement arm was employed than in either of the other two cases.

The shortest arm gave 70 oscillations, whereas the longest only gave 64 and the intermediate 66 or 68.

When the weight r was attached to g, the impulse communicated by the tooth gave 18 oscillations with the arm $a\,b$ and only 16 with the longest arm $a\,i$.

THIRD EXPERIMENT.

618.—The above will suffice to show that the greatest extent of vibration is obtained when the pressure applied is a maximum. From this it must necessarily result, if we consider a converse action, namely a recoil of the tooth occasioned by the pressure of the escapement arm, that the resistance opposed to recoil will be greatest with a short lever, providing all other dimensions remain unchanged.

Theoretically this is not open to question, but we will proceed to demonstrate it experimentally in accordance with our unvarying rule.

In order to do so, before each experiment we placed the escapement arm in equilibrium in the position indicated by fig. 1, plate VI., that is to say the middle points of the pallets were in a vertical line, and then, after drawing the arm $v\,b\,d$ backwards to the requisite distance, the tooth n was placed successively against the extremity of each incline. The arm was caused to move from left to right by a weight in the balance-pan r suspended by a thread which, after passing over a pulley y, was fixed to g.

The incline $b\,c$ moved the tooth when a weight of 20 grammes was applied.

The incline $i\,j$ required a weight of 35 grammes.

The arms $a\,n$ and $a\,s$ are to each other as 1 is to 3.

The forces required to set them in motion are as 20 : 35 or 1·71 : 3. If the pressure were the same on the two planes these forces would be exactly as 1 is to 3.

It is thus demonstrated that the friction is more intense with the plane $b\,c$. This result, which we foresaw and could easily have been deduced from the law of the inclined plane, proves that the increased pressure is favourable to the movement of the lever when this is impelled forward by the tooth, whereas, when the pallet forces the wheel backwards, the excess of pressure becomes an obstacle to the motion of this pallet.

Note on the last Experiment.

619.—The result of this experiment should be carefully noted by watchmakers.

Very many mistakes, that are accepted as truths in the workshops and in the great majority of books treating of Horology, result from the application of the false rule which asserts the extent of the acting surface to be a measure of the amount of friction.

When only the *extent* of this friction is taken into account any estimate of its value will in most cases be an exceedingly rough approximation. It is impossible to ascertain the force absorbed by friction if its *intensity*, that is to say the pressure, is not determined, or, when this is variable, the several pressures under which friction occurs (**37**).

We have just seen a striking instance of this; a much greater amount of force was absorbed by *b c* notwithstanding that its surface is considerably less than that of *i j*.

RESISTANCE TO UNLOCKINGS.

Unlocking without Recoil.

620.—When the wheel rests on a surface concentric with the axis of rotation of the locking piece, the resistance at the unlocking is proportional to the radius of rest.

It would be only necessary to repeat here the demonstration given in article **235** and the following: we therefore refer the reader to it,

Since, when the rest occurs on such a concentric surface, the pressure is uniform, the energy lost by the balance during the unlocking is proportional to the extent of acting surface. This loss of energy in a detached escapement is unimportant, but if the resting point is not absolutely stationary the resistance to unlocking is variable.

621.—While laying down this principle, which, in the present instance at least, hardly seems to require proof, namely, the principle that the balance loses more energy of motion according as the period during which it is opposed by the pressure of the wheel during rest is the more prolonged, we would point out that we hardly insisted upon this enough when discussing the escapements with frictional rest.

In that case, with a definite impelling force, the momentum lost by the moderator varies with the extent of friction during rest. Hence it necessarily follows that the movement of the escapement becomes more and more sluggish as the extent of this friction is increased.

Unlocking with Recoil.

622.—The resistance opposing the motion of the balance when the unlocking occurs is, with a straight locking plane, dependent on (1) the inclination of the plane; (2) the extent of its acting surface; (3) the virtual length of the lever by which it is set in motion; (4) the velocity of translation of the plane; and (5) the direction of the pressure.

Let us first consider two cases in one and the same escapement where all the conditions remain constant except two: the inclination of the plane and the depth of the wheel. In the first of these the lengths of the frictional surfaces remain the same but the inclinations are different; hence the amount of recoil will vary.

In the second the extent of this surface varies for the same angular movement of the escape-wheel backwards; the recoil, then, is invariable.

Inclined locking faces of the same acting length.

623.—In fig. 2, plate VI., consider a tooth of the escape-wheel as first acting at the point *d* and afterwards at *f*.

The extreme length of the arm of rest is *b* M; let this be 110 millimetres.

Assume that *b d* is equal to *b f*, and the arc *c d* is half the arc *c f*. Hence it follows that the recoil due to the plane *b f* is double of that which results from the displacement of *b d*.

By construction (neglecting small fractions) we have:

M *b*	equals	110	millimetres
g k	,,	102	,,
i k	,,	100	,,
a d	,,	45	,,
n f	,,	43	,,
b n	,,	20	,,
b a	,,	15	,,

It is almost unnecessary to observe that the extreme radius of rest *b* M remains constant, since its fixed point M is unchanged whatever be the inclination of the plane, whereas the virtual power lever (that is, the distance between the centre of movement and the middle point of the acting inclined surface under

consideration), which gives a measure of the force effecting the translation of the plane, varies with each re-adjustment of the inclination.

In the cases we are considering the two virtual levers $g\ k$ and $i\ k$ are to each other as 102 : 100; the force exerted is inversely as these numbers.

Hence the force acting on $b\ f$ is slightly in excess.

As will be evident, this excess is almost inappreciable; so with a view to simplify the succeeding argument we will, for the present, assume them to be equal and endeavour to ascertain the amount of resistance to be overcome.

Taking the two planes $b\ d$, $b\ f$, and dividing their base by their height, we have:

$$\frac{a\ d}{a\ b} \text{ or } \frac{45}{15} = 3{\cdot}00.$$

$$\frac{n\ f}{n\ b} \text{ or } \frac{43}{20} = 2{\cdot}15.$$

So that, when the planes are impelled with the same force, if one, $b\ d$, is capable of overcoming a resistance of 300 grammes, the other $b\ f$ can only counteract a resistance of 215 grammes, or just over two-thirds the first amount.

624.—We have assumed the two forces to be equal, but that acting on $b\ f$ is known to be slightly the greater. This difference of about $\frac{1}{50}$th the total amount will only very slightly influence the result obtained and the conclusions we shall draw from it; as in the preceding part, moreover, we shall continue to deal with round numbers exclusively. If it were our intention to give accurate figures it would be essential to take account of minute differences in the velocity of translation, etc., etc., and this would involve elaborate calculations with no corresponding advantage, for we only require to demonstrate that:

While retaining the same extent of acting surface, if the inclination of the plane that produces the recoil be so far altered as to occasion twice the amount of backward movement, the motive force will require to be increased to nearly a third as much again;

And it must be further observed that:

Since the angle $d\ b\ n$ is 71° and $f\ b\ n$ is 65°, the difference between the two inclinations is little more than one-thirteenth of the angle $f\ b\ n$, whereas the difference between the two forces

that impel *b d* and *b f* amounts to at least five-thirteenths of the lesser force.

These two propositions being granted we may conclude that:

625.—*The resistance due to recoil* (when the length of acting surface remains unaltered) *will vary inversely with the inclination of the plane producing recoil, but its increase will be much more rapid than the corresponding decrease in this inclination.*

For the present this is all we need prove.

Inclines producing equal recoil.

626.—Let us compare the resistances opposed to two straight inclined planes with acting surfaces of unequal length, when each communicates the same angular backward motion to the wheel.

Assume *f g* (fig. 3, plate VI.), the interval between the two concentric circles *f* M, *g a*, to accurately represent the amount of recoil of the wheel due to the displacement of the two planes M *g*, M *n*, from *a* towards *g*.

The two resting points are at *n* and *g*.

The points of application of the force are at *b* for the plane M *n* and at *a* for the plane M *g*.

P M is the extreme radius of rest.

Hence, neglecting fractions, we have:

P M	is 80	millimetres.
P *b*	,, 70	,,
P *a*	,, 67	,,
d g	,, 36	,,
d M	,, 24	,,
c n	,, 19	,,
c M	,, 17	,,

Dividing the base *d g* by the height *d* M, and the base *c n* by the height *c* M;

$$\frac{d\,g}{d\,\mathrm{M}} = \frac{36}{24} = 1{\cdot}50$$

$$\frac{c\,n}{c\,\mathrm{M}} = \frac{19}{17} = 1{\cdot}11$$

From which it is evident that, when acted on by the same force with its point of application at *a* or *b*, the plane M *g* would

overcome a resistance of 150 grammes, whereas the other M *n* would only correspond to 111 grammes.

But the lengths of the two virtual levers P *a*, P *b*, are to each other approximately as 22 is to 23, so that the plane M *g* will neutralize a force somewhat in excess of the calculated amount.

The angle *g* M *d* is 56° and *n* M *c* is 50°; these inclinations differ by about one-tenth, whereas the forces that would set them in motion differ by nearly one-third.

627.—Here, as with unequal recoils (**625**), the resistance due to recoil with the short plane increases much more rapidly than the inclination of this plane diminishes; but it would appear that the force necessary to overcome this resistance must be reduced by two circumstances that are disregarded in the illustration of article **623**.

We must, in short, take account of:

(1) The diminution in the acting surface; for M *n* is little more than half of M *g*;

(2) The comparative smallness of the angular motion of the plane M *n*; for it is represented by M P *s*, an angle which is exactly half of M P *f* or the angular movement of the plane M *g*.

Hence it follows that the geometrically calculated resistance of the plane M *n* exceeds the real:

(1) By about half the force converted into friction on *g* M; and,

(2) By about half the force absorbed when the duration of action is, approximately, doubled.

It is quite unnecessary to demonstrate the general proposition that the momentum of an oscillating moderator, whether pendulum or annular balance, will fall off to a less extent according as the period during which it remains in contact with any obstacle pressing on its axis is reduced, and as the frictional surface is reduced.

628.—To sum up:—When the motive force and the angle of recoil of the wheel are given, a reduction in the extent of the acting surface of the plane will, under the given conditions, produce

An *increase* in (1) the resistance opposed to the plane; and (2) the length of the virtual lever impelling this plane. Both these will cause a loss of energy. Also

A *diminution* in (1) the extent of acting surface; and (2) the

duration of the action. These effects will tend to maintain the energy of the moderator unimpaired.

By properly combining these elements it will be possible to ensure that the increase in the steadiness of the locking part is proportionately greater than that of the force.

A glance at the lines *g y*, *n z* (fig 3), indicating the direction of the pressures, will prove the truth of this fact; but, after all, the advantage secured is of little moment. The existence of *draw* it is true is more certain with the incline M *n* than with M *g*, but the displacement of M *n* requires a rather heavier moderator and the friction will therefore be the more intense, etc.

We have then not yet arrived at the final solution of the problem, but we shall shortly do so.

Draw.

629.—At the outset let us carefully explain the reason for so arranging the resting planes that they cause the wheel to recoil on unlocking.

In other words, what is the object of ***draw*** (called in French *tirage*)?

Merely to secure the steadiness of the part, whether it be lever pallet or detent, on which the escape-wheel rests after having given an impulse to the balance, and, while holding the mobile against its banking, to prevent any displacement save during the lift.

If it were otherwise, besides the danger of interfering contacts, the wheel would fall against varying points on the locking part, and the resistance to unlocking would become still more irregular.

The necessity of draw being granted, what conditions should it satisfy?

Clearly the two following:

It should secure constancy in the position assumed by the locking part;

And it should reduce the force required for unlocking to a minimum.

Hence if it be found that several inclines satisfy the first of these conditions to the same extent, we must select that plane which presents the least surface of friction and requires less force to accomplish the unlocking than any other.

The plane which ensures a locking and offers a minimum resistance to the moderator.

630.—Consider again fig. 3, plate VI., and let us assume that the plane M *g*, when pressed upon by a tooth at *g*, secures perfect stability of the locking part of the escapement.

The demonstration would be identical were we to take the plane M *n*, and we merely select M *g* in order to avoid the confusion which would otherwise occur amongst the lines.

The centre of rotation of the tooth acting at the point *g* will be at *o*, and *g x* will represent the direction of pressure. Assume the line *g x* to represent this force in magnitude as well as in direction; it must, then, be the diagonal of a parallelogram *g v x i*, in which *v x* gives a measure of the amount of force tending to cause a rotation round the axis P, in other words, the force by which the locking part is held against its banking.

Now move the centre of the wheel from *o* to *o'*. The circumference will be indicated by the circular arc *t h*. On this arc take a point *h* to the left of M *g*; join *h* M, *h o'*, and draw *h y* perpendicular to the extremity of the radius *o' h*.

The plane M *h* thus obtained will satisfy the conditions of the proposition.

For *h y* (which we may take to be equal to *g x*, notwithstanding a slight change in the friction at the pivots of the escape-wheel) gives a measure of the force both in magnitude and direction and is the diagonal of a parallelogram *h* P *y r*. The sides P *y*, P *h*, are proportional to the two resolved parts of this force. But it is evident that this ratio is practically the same as that existing between *v g* and *v x*.

Hence we see that the steadiness of the locking part is guaranteed just as well with the tooth at *h* as at *g*. As regards this point, then, we may consider the two conclusions to be identical, but there is, on the whole, an advantage in using the plane M *h*, since it requires a less unlocking force.

The demonstration of this simple fact we leave to the reader; we have already said enough to enable him to prove it from theoretical considerations.

631.—Hence, *if any locking plane be specified, it is possible, by reducing the surface of friction and adopting a certain inclination less than the first, to fix upon a new locking plane possessing all the advantages of the initial plane and, at the same time, involving the*

expenditure of less force in the unlocking. (It will, however, be necessary to move one of the centres of rotation.)

Where then is the search to end? It is impossible to fix a limit *a priori;* all we can at present say is that *there is an advantage in reducing the amount of acting surface, under the above-mentioned conditions, so long as the requisite freedom of the pivots is maintained and an absolute certainty in the several workings is guaranteed.*

Note.—One consequence of these considerations is that the locking might be caused to take place against an element of a circular arc, the steadiness of the part being unimpaired; but in that case the point of application of the pressure would fall very short of the tangential position, and it is well known that the usual argument in favour of a circular form concentric with the axis is based on the assertion that such a system enables us to set the escapement tangentially with ease.

EXPERIMENTS.

FIRST EXPERIMENT.

632.—I have constructed the apparatus shown in fig. 4, plate VI. It consists of a lever *b d* supported on pivots at *c* and carrying at *b n* a steel sector A which can be rotated if required on a fixed point *n*, and fixed in any desired position by a clamping screw. The inclination of the straight face *a n* is thus variable. The zero point of the scale is given by the chord *n a* of the circular arc *n j a* described from the centre of rotation *c*.

The face *b n* is curved, so that by inverting the sector this face becomes concentric with the centre of movement of the lever.

After balancing the lever by means of small washers held on *s*, the extremity *j* of an escapement arm (corresponding to the escape-wheel tooth), impelled by a weight, is brought in contact with the face *a n*, and a small balance-pan attached to *d* is loaded until the arm *c d* leaves the stop *k* and overcomes the resistance opposed by the tooth *j;* the weight of *f* must be taken into account.

We have obtained the following results with this instrument: the pressure of the tooth *j* was produced by a weight of 100 grammes acting at a distance of 17 millimetres from the centre of rotation.

The pan *f* was gently checked in its descent in order to ascertain the point of maximum resistance, thus finding what was the greatest amount of force necessary to disturb the equilibrium by successive trials of all the points on the surface *a n;* and, in round numbers, the release took place when the following loads were applied:

Acting Surface.	Position of the Incline.	Load.
28mm.	10°	8 grammes.
28 „	20°	12 „
28 „	30°	16 „

And when the pan was liberated immediately on being charged:

Acting Surface.	Position of the Incline.	Load.
28mm.	10°	5 grammes.
28 „	20°	8 „
28 „	30°	11 „

The zero point, from which was measured the position of the incline, was the chord *a n* of the arc *a j n* struck from the centre; measuring the angle enclosed between the inclined plane and the line *n c* in the several cases, we have:

The inclination of 10° corresponds to an angle of 65°.
„ „ 20° „ „ „ 55°.
„ „ 30° „ „ „ 45°.

It thus becomes evident that the resistance to unlocking increases in a far more rapid ratio than the angle of inclination to the radius diminishes (**625**).

SECOND EXPERIMENT

633.—In the experiments we now proceed to discuss, the amount of the pitching of the wheel with the incline was altered by moving the axis of rotation of the lever, whose extremity is seen at *j*, up or down in a vertical direction.

The pan was gently checked in its descent and the unlocking occurred with the following weights applied:

Acting Surface.	Position of the Incline.	Load.
28mm.	10°	8 grammes.
14 „	20°	10 „
7 „	30°	13 „

Or, setting the pan free at the instant of charging:

Acting Surface.	Position of the Incline.	Load.
28mm.	10°	5 grammes.
14 „	20°	9 „
7 „	30°	12 „

From the preceding article we know that

An inclination of 10° corresponds to an angle of 65°
„ „ 20° „ „ „ 55°
„ „ 30° „ „ „ 45°.

And from these last experiments we see that, not only the resistance increases much more rapidly than the angle decreases, as in the former case, but the advantage secured by reducing the frictional surface and the angle of inclination is more than counteracted by the resistance opposed by a virtual lever that is a little longer at the outset (**628**).

THIRD EXPERIMENT.

634.—This was made in accordance with the conditions indicated in fig. 3, plate VI., only considering the planes M *g*, M *h*. The centre of the wheel required, therefore, to be displaced in a vertical direction.

Acting Surface about	Position of the Incline.	Load.
27mm.	18°	11·0 grammes.
15 „	10°	7·5 „

These results prove a fact of which we were already fully aware, namely that a less force is necessary to effect the unlocking when the plane M *h* is employed, but it still remains to be experimentally demonstrated that the pressure of the tooth at the point *h* secures the steadiness of the locking arm as perfectly as when the pressure acts at *g*.

FOURTH EXPERIMENT.

635.—When a locking arm is held at rest by a tooth of the wheel, it can only be displaced by a blow or by roughly shaking the mechanism. Remembering this fact, we experimented as follows:

When the lever *b d* (fig. 4) had been set in equipoise on its axis, the tooth *j* was brought to act against the face *n a* successively under the conditions indicated by M *h* and M *g* in fig. 3. The pan *f* was now charged with gradually increasing weights, which however were never sufficient to cause a movement of the lever from its position of rest. After each addition

to the load the support of the apparatus was made to shake by a violent blow; at times there was only one such blow and at other times there were several in succession.

The three following combinations were obtained by vertically displacing the centre of movement of the tooth and by inclining the plane *n a* towards *b*; the first corresponds to M *g* and the third to M *h*:

1st Combination.	Acting Surface about	27mm.	Position of the Incline	18°	} Successive charges 1, 2, 3, 4 grammes.
2nd	"	"	"	20 "	"	"	10°
3rd.	"	"	"	15 "	"	"	10°

With a load of *one* gramme the shaking of the support produced no effect.

With *two* grammes there was no disturbance in the 1st and 3rd combinations. But in the 2nd the arm was just caused to come away from the stop (or banking) *k*.

With *three* grammes, the arm *c d* and *k* were, in the 1st and 3rd cases, separated by an interval of two millimetres but never more; in the 2nd case the interval amounted to three millimetres.

Finally with a load of *four* grammes, the separation amounted to 4mm. in 1 and 3 and much more than this in 2.

It is thus seen that combinations (of inclination of plane and direction of force) numbers 1 and 3 maintain equally well the steadiness of the arm *b d*.

Resistance to disturbance when the resting surfaces are concentric. We cannot better conclude this series of experiments than by ascertaining what degree of steadiness can be secured in the escapement arm when the circular form is adopted.

After inverting the sector A and fixing it in such a position that its curved face *b p n* forms an arc of the circle described with radius *c a*, the tooth *j* was brought to act at the point *a* (which is the tangential position or very approximately so), and experiments similar to those explained above were made.

The pan *f* was successively charged with 1, 2, 3, 4, decigrammes.

With a load of *one* decigramme no disturbance occurred.

With *two* decigrammes the arm *c d* moved between one and two millimetres from the stop *k*; with *three* decigrammes it moved very nearly four millimetres, and with a load of *four* decigrammes the interval exceeded seven millimetres. The resistance opposed to disturbing influences was then about *ten times less* than in the fourth experiment above.

We thus see that by giving a circular form to the resting surfaces in detached escapements, the steadiness during rest becomes much less certain than with an inclined rectilineal form.

The above applies to the case of a tangential escapement; the note at the conclusion of article **631** must not be overlooked.

The period of an oscillation will change with any variation in the length of the escapement arms.

636.—We always assume that the lifting angle and the motive force remain unchanged.

It has been shown in **235** and the following articles that by increasing the radii of resistance the friction is made greater, so that the motion of the moderator is impeded.

An analogous demonstration would serve to show that, in a detached escapement, the resistance during unlocking increases as the radius of rest is made longer, and that this case only differs from the escapements with frictional rest in the actual amount of the resistance.

Since the effect of increasing the length of the locking arm is to check the movement of the moderator, and shortening it has a converse effect, it will be necessary, as with the former class of escapements, to determine the one length of arm with which these two effects compensate one another or, at least, nearly do so, and then the moderator, when acted on by the escapement, will behave precisely as when it is isolated and oscillates freely. (This double effect cannot be secured with a balance or pendulum that is too light.)

637.—A remark of some importance, bearing on both classes of escapements, may conveniently be made here, especially as it meets an objection that has been urged.

Do we assert that the length of these arms is invariably fixed at the point thus determined upon by theory? This subject is very complicated and we shall have occasion to revert to it; but we can partly anticipate the subject and at once state that *it is*, if the freedoms are adjusted with almost absolute accuracy, the workmanship as nearly perfect as possible, and if the pressures, being greater with short arms, do not exceed the resisting power of the materials employed; and that *it is not* in the converse case: the result will be that the touching surfaces are distorted, etc., and irregularities in the rate of the mechanism necessarily follow.

638.—In the above discussion we have disregarded the presence of the balance-spring in pocket timekeepers. But we have already seen in articles **262** and **265** that the introduction of this element leads to results that are apparently contradictory. *Apparently* for, so far from indicating any error in the theory, they confirm its accuracy and prove, as we have already seen and shall also subsequently demonstrate, that, by approximating in actual practice to the theoretically correct proportions, the period during which the regulator, whether it be a balance-spring or a pendulum, maintains its perfect isochronism becomes more and more prolonged.

The useful effect of the Impulse varies with the height of the incline by which the motive force is transmitted to the moderator.

639.—It appears unnecessary to repeat the argument contained in **247** and the following articles. We will only refer the reader to it since it is sufficient to establish the accuracy of the above proposition.

We would, however, add a few words.

All our discussions there had reference to an impulse plane of practically constant length, for, if the nature of the escapement enabled us to alter this length at will, we might, in most cases, secure the same amplitude of oscillation of the moderator by compensating for any loss of force by a more prolonged action.

But we assume the reader to be sufficiently cognizant of the principles of the subject to render further remarks unnecessary.

There is only one degree of inclination of this incline that secures the greatest extent of motion with the greatest regularity.

640.—This proposition is equally applicable to the two classes of escapements we have so far considered, so that the reader need only refer to articles **251-9** for a complete proof of it.

Any change in the dimensions of the moderator involves a change in the proportion between the Impulse and Locking Arms.

641.—The movement of the moderator is occasioned by the difference between the impelling and unlocking actions. In short, it is measured by the excess of power over resistance; just as in the case of frictional rest escapements, except that the resistance has not the same value, for in one case it results almost entirely from a continuous pressure and in the other from an impact.

The demonstration contained in **260** and the following paragraphs is also applicable, with the reservations given below, to the escapements now under discussion, but the reader must mentally substitute the expression "resistance to unlocking" for "resistance caused by friction on the resting surface."

Difference between the mode of action of the Resting or Locking arms in the two classes of Escapements.

642.—We have just drawn attention to the very different character of the resistance which precedes each fresh impulse in detached and frictional rest escapements.

It will be well to further discuss these differences, as they are of very great importance.

Escapements regulated by a Pendulum.

643.—First consider the escapements used in clocks and regulators, taking Graham's as an example.

If experience did not clearly prove that with such an escapement we secure isochronism in the oscillations of the moderator, the mere fact that we have shown the possibility of causing the ratio between the impulse and the resistance due to friction on the resting surfaces to vary at will, would show that, though not easy, it is at any rate possible to prove theoretically the existence of isochronal oscillations.

644.—But such is no longer the case if, instead of Graham's, we employ a detached escapement.

Let us proceed to examine what would occur on increasing the motive force; we are suddenly brought face to face with three sources of irregularity that would effectually prevent any isochronism being established and maintained in the movements of the moderator.

First. A more efficient impulse applied to a detached pendulum will cause it to perform longer arcs of oscillation; and with such a free moderator the long arcs of vibration will be slower than the short arcs.

Second. The unlocking is accompanied by an impact which is dependent on the energy of the pendulum, varying with any changes that may occur in its amount. This pendulum, moreover, is opposed by a weight or spring whose resistance may be influenced by several causes.

Third. Finally, the augmented resistance to unlocking, erroneously regarded as balancing the increased motive power,

is not only insufficient from the outset for this purpose, but, as we shall presently show, changes with time in a manner very different from the progressive falling off in the motive force (**647**).

The resistance accompanying this unlocking is in an almost constant state of change, and any regularity that a practically invariable impelling force might secure is thus lost.

645.—Hence the following demonstrations lead us to conclude that, although a form of frictional rest escapement has been arrived at in which the powers and resistances are favourably balanced for timing under a varying motive force, the same has not been done for the detached escapement used in regulators; such an equilibrium has not, as yet, been secured in it, nor has it been possible to obtain such accurate results. The reason of this is that in this form of escapement conditions exist which increase or diminish the influence of the moderator, sometimes tending to help the escape-wheel and at other times working in opposition to it.

Escapements with the annular balance.

646.—All the above remarks would be equally applicable to the detached escapements used in portable timekeepers were it not that in the balance-spring they possess an element of regularity that is wanting in clocks. The existence of this one element changes all the conditions of the problem as we shall presently show.

The ratio between the Impelling force and the Resistance on the resting Surface changes with time.

(Applicable to all Dead-Beat Escapements.)

647.—This branch of the inquiry is entirely new. To the best of our knowledge the question has not been previously discussed by anyone, and its explanation will make evident the causes that give rise to phenomena hitherto unexplained in connection with practice and observation.

The majority of horological authors, as well as most watchmakers who have studied the subject experimentally, consider the correcting influence which the dead-beat escapement has on the inequalities of motive force to be "due to the fact that an increase of force accelerates the vibrations of the moderator, while the pressure on the locking part, or the resistance to unlocking, being also augmented, reduces this excessive velocity."

This explanation, which is incomplete and therefore erroneous, would lead one to suppose that with any variation

in the motive force the energy of impulse and the pressure on the locking face increase or decrease together just as the force itself does, but this is not the case.

Before proceeding to give a theoretical demonstration, we will, according to our usual practice, refer to certain confirmatory experimental evidence already known to our readers which would be sufficient for many of them; we are, however, anxious to leave no doubt whatever in their minds.

648.—Gradually increase the motive force of an horological appliance provided with an escapement, and a point is nearly always reached at which the mechanism stops.

It is manifest that if the energy of the impulse, which at first is of course greater than the resistance to unlocking, were to vary in proportion to this latter, no point would be reached at which a setting occurred, but there would be merely an increased rapidity of movement.

Remark.—We say that nearly always a point of setting is arrived at, but it is necessary that the balance be light rather than heavy. For a heavy balance acquires a considerable energy of movement when the force is increased, and this might break the pivots before the point was reached where the movement is paralysed by the pressure exerted on its axis.

649.—Let us now proceed to the theoretical consideration of the subject.

In order to simplify it assume the force to be successively increased fourfold.

The laws of Mechanics show that the several progressions will be as follows:

The motive force successively	1, 4, 16, 64....
,, pressure ,, ...	1, 4, 16, 64....
,, resistance to unlocking successively .	1, 4, 16, 64....
,, energy of the balance ,, .	1, 2, 4, 8....

It is thus seen that, whereas the resistance is quadrupled, the energy of the balance is only doubled. It must therefore follow that, although at first the energy of the impulse was considerably in excess of the resistance to unlocking, a point is ultimately reached where they balance, and then the motion of the moderator will cease.

650.—In following this subject more into detail we shall meet with one element of the problem of the determination of

the proper weight of balance as well as of the motive force best suited to a given mechanism.

651.—The initial ratio between the impulse and unlocking will be modified by time, since the force acting on the escapement diminishes as the oil on the pivots becomes thicker. But, theoretically, the change will be the inverse of what is indicated above, that is to say the force of unlocking will only be reduced one half when the motive force becomes less by three quarters.

We say *theoretically* because, as will be gathered from the experiments presently to be described, the figures are not absolutely correct, and the presence of this oil on the balance pivots, the resting surfaces and the pallets of the escapement modifies the proportions. This action may, in the case of timekeepers for general use and with escapements such as Graham's, amount to a species of compensation, more or less effectual according to the quality of the oil.

In the case of chronometer escapements we are not possessed of sufficiently accurate data to determine the value of this influence and its effect on the rate.

The ratio between the Impulse and Unlocking will be more or less influenced by time according to the length of balance-spring employed.

(Applicable to Dead-Beat Escapements generally.)

652.—As with the above discussion, this subject is new.

The energy of the balance in virtue of its motion may be regarded as a product of its mass into the velocity it possesses at the moment under consideration; that is to say, when the unlocking commences. Such an assumption, it may be observed, appears to us to most nearly approximate to the truth, and to be best borne out by experiments.

This velocity of the balance after the reversal of its motion is closely connected with the gradual increase in the tension of the balance-spring.

In every balance-spring, providing it is perfectly even and homogeneous throughout, there is one length which gives isochronous vibrations of the balance, that is to say each vibration, whatever its extent, occupies one and the same period of time; this fact was discovered by Pierre Le Roy. Beyond that point the long arcs require more time than the short arcs, and when any less length is taken the reverse is the case.

653.—With a view to simplify our explanations let us assume

that three lengths of a certain balance-spring when tested have given:

The *first*, long arcs half as quick as the short arcs;

The *second*, long and short arcs isochronous;

The *third*, long arcs twice as quick as the short arcs.

Represent the mass of the balance by 1 and multiply this unity by the velocity possessed by the moderator at the end of long and short arcs; we shall find that the energy increases or decreases very nearly in the following proportions:

With the 1st balance-spring as	1	:	4	: 16
,, ,, 2nd ,, ,,	4	:	2	: 1
,, ,, 3rd ,, ,,	16	:	4	: 1

That is to say, speaking generally, with the first spring the energy of the balance (as compared with the force exerted on the escape-wheel at different periods) will gradually increase with time;

With the second (isochronal) spring it will diminish in a certain ratio;

And, finally, with the third, it will diminish in a very much more rapid proportion.

654.—The length of the balance-spring is, so to speak, one factor in the timing. The great difficulty of this operation lies, not in securing a certain *rate at the time* but in making sure that this *rate shall be maintained in future.*

We were, then, justified in saying that the discussion, so often carried on between the advocates of long and short balance-springs, could lead to no useful conclusions. An escapement wants neither a long nor a short spring; in other words it does not require a spring chosen solely with a view to the extent of the arcs of vibration or the range allowed by the curb pins, but what is imperatively necessary is a balance-spring of *one definite length*, adapted to neutralize the modifications that time effects in the going of the escapement.

655.—This novel law, for we look upon it as such since the rules observed by springers for ensuring the rates of their chronometers are utterly empirical, will suffice to account in a very simple manner for the so-called peculiarities observed in the rates of chronometers. It will to a certain extent explain why at sea equal success has attended the use of chronometers, some of which were provided with perfectly isochronal springs while in others the springs occasioned very grave differences

between the long and short arcs; and, lastly, why springs are equally serviceable whether their fixed points are at the extremity of a diameter or are brought near the centre by curving the extremities, etc. It will, moreover, explain one of the circumstances that led makers to deviate from perfect isochronism in chronometer springs so that the short arcs are slightly accelerated, the practical utility of which they had been led to discover by experience.

EXPERIMENTS.

FIRST EXPERIMENT.

656.—Those of our readers who are sufficiently advanced in the mathematical sciences will be satisfied by the above reasoning. In the interest of those who are less favoured we will resort to experiment: it always yields valuable information to those who know how to employ it aright.

We will again employ the apparatus described in article **632** and represented in fig. 4, plate VI. Remove the balance-pan *f* and effect the unlocking by the fall of a weight perpendicularly on the point *s*.

Knowing the height through which the body falls we shall, by means of the law of falling bodies *in vacuo* (**126**), be able to ascertain the velocity at the moment of impact, and the law of instantaneous forces will give the energy of the blow.

The figures obtained will not be absolutely exact: since the weight is hollowed out in the form of a bell so that it may strike with its centre of gravity and remain suspended on the pointer *s*, it must necessarily follow that the air collected in the cavity slightly modifies the theoretical conditions of the fall; but the results arrived at will be sufficiently near since they are only employed to demonstrate the proposition to watchmakers who are but slightly familiar with the laws of Mechanics. Those possessed of a good theoretical knowledge will, as we have already observed, not require the evidence of these experiments.

The following results have been obtained:

When the lever *b d* was evenly poised on its pivots, the tooth *j* was placed in contact with the plane *a n* just above the point *n*, care being taken that the inclination of this plane was sufficient. It will be remembered that the pressure of the tooth can be increased at will, for it is produced by a weight suspended to a cord on the axis of *j*.

The falling weight, weighing 6 grammes, was suspended by a fine thread passing over a pulley; the fall thus took place without shake on burning this thread.

According to the law of falling bodies we should have, employing always the same weight:

Height of fall, 1;—velocity, 1;—force exerted, 1.
„ „ 4;— „ 2;— „ 4.

The results actually obtained with the apparatus were:

Height of fall, 10 mm.;—resistance overcome, 59 grammes.
„ „ 40 „ ;— „ „ 116 „

We thus see that about half the force exerted by the body at the moment of impact is without useful effect; it is dissipated by resolution, elastic reaction, friction, generation of heat, &c. The actual effect may then be approximately represented in accordance with the law of momentum (**125**).

SECOND EXPERIMENT.

657.—A wooden disc, bearing a small metal tongue projecting from its edge, was supported on an axis and so placed that when caused to rotate the tongue struck the extremity *d* of the lever arm *c d*, fig. 4.

The tongue, being placed at an angular distance of 160° from the point *d*, was driven by a non-continuous blow produced by the fall of a mass pivoted at the extremity of a lever.

Fall of this mass, 9 mm;—resistance overcome, 70 grammes.
„ „ 18 „ ;— „ „ 140 „

658.—After verifying and noting these results, a thread was coiled on the axis and a balance-pan containing a weight of 15 grammes was attached to its extremity. The pan was arrested by a support just before the tongue came into contact with *d*.

By moving the tongue backwards first through an angle of about 82° and subsequently through 330° the pan was made to fall through a distance of 8 millimetres in the first instance and 32 in the second.

Height of fall, 8;—resistance overcome, 35 grammes.
„ „ 32;— „ „ 140 „

Here, then, the useful effect is proportional to the angular

path of the striking body and to the mass set in motion, that is the disc. These results tend to confirm the opinion that, practically, the energy of a balance or its useful effect is most nearly expressed by its momentum.

THIRD EXPERIMENT.

659.—We would, at the outset, observe that when the balance turns under the influence of the escape-wheel its power, as measured by the impact it would be capable of producing, depends on the energy of the impulse; but during the return vibration its energy results solely from the influence of the balance-spring; this again varies in accordance with the amplitude of the angular movement of the moderator.

660.—The wooden disc carrying a metal tongue was replaced by a heavy balance, about 8 c.m. (3·15 ins.) in diameter, supported on fine pivots and provided with an isochronal balance-spring.

The pin was at *x*, beneath the rounded corner of the arm *d*, fig. 4, plate VI., when the spring occupied its neutral position; so that by moving the balance in the direction *d r*, the balance-spring was coiled up; the moderator, when released, rotated in the direction of the arrow, and the pin struck the arm *d*, unlocking the resting plane *n a* from the tooth *j*, providing its energy was sufficient. It will be remembered that the pressure exerted by this tooth is produced by a weight that can be changed as required.

The arc of lead, that is the arc through which contact occurred between the pin and lever arm, was 26·25°. If this amount be subtracted from the total arc of vibration, we know the extent of the free or supplementary arcs.

We shall only consider a semi-vibration, that is the arc traversed on one side of the point of rest.

Free Arc.	Total Arc.	Resistance Overcome.
3·75°	30·00°	30 grammes.
7·50°	33·75°	60 ,,
11·25°	37·50°	90 ,,
15·00°	41·25°	120 ,,
18·75°	45·00°	150 ,,
22·50°	48·75°	180 ,,
26·25°	52·50°	210 ,,
30·00°	56·25°	240 ,,
33·75°	60·00°	270 ,,
37·50°	63·75°	300 ,,

661.—The figures given in this table confirm, in a remarkable manner, our theoretical conclusions: for it will be noticed that:

(1) It is the formula for the momentum of a moving body that gives the nearest approximation to the energy of the moderator, regarding it at the commencement of the unlocking. Thus the free arc of 37·5°, which is double the arc 18·75°, overcomes twice the resistance; and the arc of oscillation 15° is four times the arc 3·75°, and is capable of overcoming a weight four times as great, etc. (see **1309**).

(2) The progressing force of an isochronal balance-spring is perfectly uniform and in accordance with the law enunciated in M. Phillips' *Mémoire sur le spiral réglant:* a law which asserts that, so long as no distortion occurs, the force exerted by the balance-spring is proportional to the angle through which the balance is moved. We thus see that when this angle is doubled, trebled, or quadrupled, etc., the moderator will overcome twice, thrice, etc., the resistance: say, 120 grammes for every arc of 15°.

FOURTH EXPERIMENT.

662.—Having obtained a measure of the energy of the moderator when brought back by the balance-spring, we now require to know the force that sets it in motion, and the manner in which this force should increase or decrease so as to cause the balance to perform the arcs indicated above when impelled by the escapement arm.

This problem is somewhat difficult of solution whether we resort to calculation or experiment; for with a given moderator united to a balance-spring, identical forces will, if the conditions under which the lift occurs be slightly varied, produce a succession of supplementary arcs varying in extent.

Remembering these facts and only considering the case we have here discussed, the following is the manner in which this final experimental verification has been made and the results arrived at are also given.

663.—I first arranged a straight lever, revolving on a pivoted axis so that one of its arms might be acted upon by the pin of the balance while the other was charged with a variable weight. The latter arm was supported by a thread, and when this was released a rocking motion was imparted to the lever by the weight; the balance thus received an impulse and the lifting arc measured 22·5°.

I next repeated the experiment employing another lever

with arms of different lengths that only acted on the balance through a lifting arc of 15°.

664.—First Lever.—Lift, 22·5°.

Weights Employed.			Arcs Traversed.
15	grammes	... ...	30·00°
30	,,	... ...	41·25°
50	,,	... ...	52·50°
80	,,	... ...	63·75°
125	,,	... ...	75·00°

665.—Second Lever.—Lift, 15°.

10	grammes	... ...	30·00°
23	,,	... ...	41·25°
76	,,	... ...	52·50°
390	,,	... under	63·75°

The lifting arc is included in these angles, for what we require to ascertain is the distance traversed by the mark on the balance rim from the neutral point of the balance-spring; in other words, we are considering the angular movement of the balance from its position of rest.

666.—An examination of these two tables shows that: (1) In order to double the energy of a moderator when returning to accomplish the unlocking, it must have been acted on by more than four times the force, because, in addition to the resistance opposed by the mass to its motion, we have to take account of the increasing resistance of the balance-spring (**651**). (2) The rate of increase in the motive force (or its decrease with time) not only differ from the rate of augmentation or diminution in the energy of the balance, but the extreme limits of its variations are dependent both on the extent of the lift and on the conditions under which it occurs.

Experience confirms theory in this case also.

To Determine the Size of the Escape-wheel.

667.—As has been shown in **266** and the following articles, the size of the escape-wheel is not a matter that can be left to chance. We need only refer to the discussion there given of the subject, supplementing it by the following additional observations.

668.—Assuming the size of the moderator, whether it is a pendulum or annular balance, to have been determined so as to correspond with the number of vibrations it is required to

beat, the size of wheel suitable in the generality of cases is determined by the escapement arms being of such dimensions that the moderator when under their influence, that is when engaged with the escapement, will behave as nearly as possible as though it were detached.

This observation must be taken in conjunction with article **637**, and it must further be remembered that any modification in the diameter of the wheel or the size of its teeth may change the initial ratio between the impulse and the unlocking. We have already seen the consequences that result from this.

669.—A small wheel exerts too severe pressure, whereas a large wheel is checked by the thickening of oil.

The velocity is of less importance than the conditions determining the lift and the pressures by which it is accompanied. An excessive velocity, especially at the beginning of the lift, increases the pressure on the plane or the impulse pallet. Too much velocity towards the end makes the severity of the drop excessive (**672**).

CONCLUDING OBSERVATIONS.

670.—The reader would do well at this stage to again study with care articles **269** to **272** inclusive, selecting those portions that are applicable to the detached escapement. The following notes will serve to complete this selection.

Summary of the latter propositions.

671.—Several consequences follow from the propositions just demonstrated.

(1) When employed in portable timekeepers, the detached escapement will not possess a reliable rate unless provided with a suitable isochronal balance-spring.

(2) In clocks and regulators its performance has been indifferent because it is influenced by any inequalities in the motive force, either during the impulse or the unlocking; and we have no force available that is constant or at least sufficiently so.

(3) With a given motive force, any change in the conditions of the draw on the locking surface will cause the rate of the escapement to vary in time.

(4) Absolute isochronism can only exist in theory. Such isochronism as can be attained to in machines for the measurement of time is always relative and is modified by any changes that may be occasioned in the future by time.

The Lift.

672.—We refer the reader to article **272** and must here again insist on the utter rejection of every vestige of that false theory which required the lift to be *equal;* by this expression we understand that, if the acting surfaces of the wheel and impulse pallet be subdivided into the same number of equal parts, each portion of the wheel should, during the lifting action, displace the corresponding equal part of the pallet.

We have already said, but it will, nevertheless, be well to repeat it here, that there is no definite rule to be followed, and the watchmaker must ascertain by calculation and experience which of the three kinds of movement:—uniformly accelerated; uniform; or uniformly retarded motion;—is best suited; that is to say produces vibrations of the requisite extent, occasions the fewest irregularities in the locking action of the escapement and requires the least motive force; in other words, a force that does not occasion wear at the points of contact.

Speaking generally, the nature of the lift which seems best to secure a uniform rate is determined by *so co-ordinating the lifting angle and the forms of surfaces of contact* that there is the least possible difference between the two actions of the motor that are required to produce the great and the small arcs of vibration (**666**—2).

The Drop.

673.—Notwithstanding that the drop of the tooth on to the locking piece has no direct influence on the balance in the detached escapement, yet, when this drop becomes considerable, it cannot fail to have a detrimental effect in the shake that is imparted to the entire system; this shake is apt to cause a change in the position of the point of contact owing to the rebound of the locking piece (pallets or detent), or the vibratory movement imparted to it by the impact.

Conclusion.

674.—Henceforth then we are justified in considering it to be satisfactorily established that constancy in the rate of a detached escapement is secured by correctly combining those effects or, as they may be called, terms, that are modified by time; they must be in such accord with the length and degree of isochronism of the balance-spring that this latter tends to neutralize their variations.

It necessarily follows that the desired end may be attained by many combinations and proportions, providing that the length and isochronism of the balance-spring are in accordance

with the demands of observation and correspond to the changes that are brought about by time. (See the Articles on *Springing* and *Timing* in the Third Portion of this work.)

The above remark indicates the cause of the great diversity of opinion expressed by watchmakers who are engaged on the better class of watches and chronometers (**655**), and at the same time it explains the excellent rates of the chronometers by Harrison and the two first by F. Berthoud. They gave the longitude with sufficient accuracy at sea notwithstanding that their balance-springs were not isochronal, at any rate in the sense in which we now use the word. These successes might have indicated even in their day that isochronism is not the only basis of timing, and evidence of this fact has since been adduced.

675.—And yet we must add, in conclusion, that it appears to follow from careful observations, which however are few in number, that the rate of a chronometer becomes more and more perfect and will be the more nearly the same after each cleaning of the instrument, according as the escapement permits the use of *a more perfectly isochronous balance-spring.*

Pierre Le Roy appears to have foreseen this relation between the perfectibility of timing and the close approximation of the actual isochronism of the balance-spring to absolute isochronism; and we would here remark that the day is perhaps not far distant when horologists will acknowledge how hopeful was the branch of inquiry opened up by the genius of this great man, and how much more fruitful of results it might have been if only followed up.

On the tangential rest and impulse in Detached Escapements.

676.—Our earlier studies and the above theoretical and experimental demonstrations suffice to prove that there cannot be any condition so absolute as the so-called rule for making either the impulse or rest tangential.

Any advantage attainable by such an arrangement must be considered secondary; that is to say it is of less importance than those requirements to which we must necessarily conform, and which, in the majority of cases, prove the tangential escapement to be an illusion.

We will, then, merely draw attention to this subject here, and we shall consider it in detail when determining the conditions that secure a proper impulse and steadiness of the locking arm in lever and chronometer escapements.

LEVER ESCAPEMENT.

CHAPTER I.

Preliminary.

677.—The lever escapement employed in watches is derived from the anchor dead-beat escapement used in clocks, and called Graham's escapement after its inventor. In order to apply this latter escapement which only allows of very small arcs of vibration, to the watch, it was necessary not only to alter its form but also to make the balance independent of the motive force except during the actual period of lift.

Thomas Mudge, a clever English watchmaker, was the first to satisfy these requirements, by an escapement in which the two lifts were equal and an impulse was given at each vibration of the balance. This he made prior to 1770, but a correct description was only published by his son in 1799, several years after his death. Robin, a French watchmaker, gave a description of the escapement that bears his name in 1792; in it each alternate vibration is *dumb*.

Finally in the museum of the Society of Arts of Geneva may be seen a model of an escapement constructed in 1786 by Pouzait, a watchmaker of that city. In it the anchor is independent of the action of the balance during the supplementary arc.

The modern lever escapement in which the pallets and pivot-holes are provided with jewels, may, if well made in conformity with the principles of Mechanics, be considered to be the best adapted for ordinary use. It will go for a long period without wear of the acting surfaces, only requiring simple cleaning from time to time and the renewal of oil when necessary. Its rate is equal to that obtainable from well-made duplex escapements, except in the case of carriage clocks, etc., where it is less satisfactory; and amongst other advantages it has over the duplex, it is more solid, allows of arcs of vibration in excess of 360°, and does not require such absolute accuracy in its construction, although at the same time it must not be carelessly made.

In spite of the excellent results that have been obtained numberless times with this escapement, which we cannot think to be due to the workmen, for the most part ignorant, who construct and repair it, and although its peculiarities have been known almost since its invention, it has been but ill understood by a large number of watchmakers who have either not sufficiently experimented on the subject or have derived such knowledge as they possess from badly constructed escapements.

Neither Antide Janvier nor Louis Berthoud seems to have thought much of this construction, a fact which may be partially accounted for, since the first of these occupied himself entirely with orreries, and the second with marine timekeepers, where the lever escapement, except in very rare instances, is unsatisfactory. But it is not so easy to account for certain modern ideas on the subject and for a published description of the several escapements mentioning the lever as inferior to the verge.

It is only fair to add that, although the rate of the lever is equal to that of the duplex, it is not maintained for so long a period. The former is soon affected by oil, which has hardly any influence on the latter; it is thus no uncommon thing to meet with duplex watches that have gone for three years in all climates without appreciable variation, whereas we rarely see a lever watch maintain its rate for more than eighteen months at a time. The need of more frequent cleaning is not a sufficient reason for rejecting so excellent an arrangement, and we have already, while discussing the duplex, given reasons for prefering the lever escapement. It is possible, moreover, to plan it so as to be less sensitive to the thickening of oil than is usually found to be the case.

Denomination of the Several Parts.

678.—This escapement consists of:

(1) The balance-staff. On this staff a steel disc (▲, figs. 1 and 2, plate VII.), which is known as either the *roller* or *table*, is firmly held by friction. This roller carries a small ruby-pin (*o*, figs. 1 and 2) projecting downwards perpendicular to its surface; this is known as the *ruby-pin* or the *unlocking pin*.

The staff itself (filled in by dotted lines *h t*) and the mode in which the roller is fixed are shown at *f g*, fig. 1.

(2) A pair of pallets movable on an axis and consisting of two arms or levers usually provided with ruby (E, G, figs. 1 and 2). These are distinguished as the *engaging pallet*, E, and the *disengaging pallet*, G.

The parts *m a*, *n c*, of the pallets are known as the *locking faces;* and *a b*, *c d*, are the *impulse planes*, *inclined planes* or simply *inclines.*

The *fork* or *lever*, which is merely a prolongation of the pallets towards F, or, as in English watches, a separate piece fixed to them, has two arms known as the *horns* or *prongs* I, J; between these two and above the slot separating them is, in some Swiss watches, a small prismatic projection F, generally termed the *dart*, which is replaced in English lever escapements by a thin pin starting from the flat of the lever and called the *guard-pin.*

The portion *k* is merely to balance the fork so as to put the whole in equipoise.

(3) A flat escape-wheel whose teeth are sometimes pointed and at other times clubbed as indicated in fig. 2. In the latter case the extremity is formed into a short inclined plane.

Action of the Escapement.

679.—When the mainspring of the watch is let down and the balance at rest with the balance-spring in its neutral position, the pallets, having no tendency to move, will remain in the position indicated in fig. 2, plate VII., being held there by the ruby-pin in the fork of the lever.

Assume the tooth that first comes into action to be at H′, fig. 2.

On winding up the mainspring, the wheel at once moves, turning towards the right. The tooth H′ is driven to H, advances along the incline *a b*, and, after forcing it backwards sufficiently, escapes as shown at N, fig. 3. This completes a half-lift, and the wheel falls against the locking face of the pallet G where the tooth is held (fig. 3).

The motion of the pallet caused by the passage of the tooth along the incline is transmitted to the balance by the fork and roller action. The side *r* of the notch, pressing against the ruby-pin *o*, drives it forward so that it escapes from the notch. A rotary movement is thus imparted to the balance which is rigidly connected with the ruby-pin.

So long as the balance continues in motion entirely disconnected from the rest of the escapement, the lever remains in contact with the banking K (fig. 2), being held in that position by the pressure of the tooth M against the locking face of G (fig. 3).

The balance under the influence of the balance-spring is brought back on its path, the pin enters the notch of the fork and drives the lever with it in consequence of the momentum possessed by the balance.

The locking of the wheel now terminates because the tooth M (fig. 3), on reaching the edge of the incline *x z*, impels it forward, at the same time travelling along its surface. This pressure, accelerating the motion of the lever, causes the velocity to suddenly exceed that of the roller by which it is impelled; forcing this forward in turn, it restores to the balance the energy necessary to maintain its vibration, and this is completed, as in the former half, in entire independence of the other parts of the escapement. After being brought back by the balance-spring it accomplishes a fresh unlocking, and so on through all the succeeding vibrations.

Two distinct effects are produced by the wheel; it performs the locking action, that is to say it causes a momentary stoppage of the train, and transmits the motive force to the pallets.

The action of the fork may also be divided into two portions, the one active and the other passive: it is passive when impelled by the roller until the wheel is on the point of being unlocked, and active when driving the ruby-pin and thus transmitting to the balance the force exerted by the wheel.

Functions of the Roller and Guard-Pin.

680.—An examination of fig. 3, plate VIII., shows that the horns of the fork are of no service in the mere action of the escapement, for the ruby-pin must pass them without contact, being only required to touch the sides of the notch between them. They are, however, useful in preventing the escapement from failing in its action through possible errors in its construction.

Their external edges serve as a means of avoiding over-banking, since the pin will strike against them when the extent of the vibrations is excessive (A, fig. 3).

The path of the lever, that is its angular movement between the two lockings, is limited either by two bankings

(Y, Y', fig. 1), fixed in the plate of the watch, or by the corners of the hollow in which the lever works in some Swiss watches.

Notwithstanding that the lever is held against the banking by the pressure of the tooth of the wheel on the locking face, it might possibly be displaced by a shake, and the action of the escapement would then be faulty. Such an accident is prevented by leaving a small triangular prism on the fork at C (figs. 3 and 4) termed the *dart*, which is replaced, in the English form of lever escapement, by the *guard-pin*. This renders over-banking impossible. Should any displacement occur, it will at once press against the edge of the roller and retain the fork in such a position that the ruby-pin may pass into the notch (fig. 3, plate VIII.).

The entrance of the dart or guard-pin into the passing hollow B (figs. 3 and 4) is coincident with that of the ruby-pin into the lever notch, an arrangement which prevents the roller from interfering with the transfer of the lever from one to the other banking.

It will be evident that the ruby-pin and guard-pin remain in the lever notch and passing hollow respectively during the same period.

Any actual contact between this guard-pin and the roller can only be caused by an accident. A slight freedom is allowed between them.

This explanation will suffice to show that, in addition to being very safe in its action, the lever escapement possesses the two great advantages of long arcs of vibration and a non-liability to set.

VARIETIES OF CONSTRUCTION.

681.—The form of escapement represented in plate VII. is known as a *right-angle* escapement.

In some of the best watches another arrangement is adopted; it is known as the *straight line* escapement because the three centres of the wheel, the pallets and the balance are in a line. The one shown in figs. 1 and 2, plate VIII. is of this class.

The portion x x' is merely added in order to put the lever in equipoise; it balances the fork end.

682.—In escapements of this form the ruby-pin is usually

fixed at one end of a small lever z' B z, and the ordinary roller is replaced by a small or *safety roller*, h (figs. 1 and 2), whose diameter is considerably less than that of the small lever.

A pointed index-piece r is screwed to the under side of the fork and answers the purpose of the guard-pin; when the ruby-pin enters the fork this stands opposite to a small notch formed in the safety roller.

This form of escapement, known as the *double roller*, is much more delicate than that represented at figs. 1 and 2, plate VII., and requires very great accuracy of construction, but it has the advantage of being more certain in its action, and when the index-piece does accidentally come in contact with the edge of the roller, the friction occasioned is less severe than that which occurs between the guard-pin and the edge of the table-roller.

Moreover it reduces the extent of the possible displacement of the pallets when a shake occurs, the penetration in the roller notch (*c d*, *n m*, fig. 4, plate VIII.) being greater with the circle of less diameter.

The details that are omitted in these figures can easily be supplied by comparing figs. 1 and 2 of plate VIII.

Banking against the Escape-wheel Axis.

683.—In some beautiful escapements of Swiss manufacture, the angular movement of the lever is limited, not by two banking pins (Y, Y', fig. 1, plate VIII.), but by the axis of the escape-wheel, against which one or the other arm of the piece x x', a simple prolongation of the fork, rests; this prolongation is a suitably formed piece of steel centred on the pallet-staff and fixed by a screw.

This appendage, serving to balance the fork, was formed as above explained with a view to avoid adhesion between the fork and its banking pins, for the point of contact with the axis is constantly changing.

When this construction is adopted, although the wheel remains perfectly stationary during the period that its axis is pressed upon by either arm, it is well to make the axis no thicker than is required to secure solidity, in order to diminish as much as possible the extent of the surfaces in contact.

At the present day, escapement makers, while adding the appendage x x' to the lever, yet set the arms so far apart that

they cannot come in contact with the axis of the wheel; the fork then banks against the pins Y Y'. Is this done because less minute accuracy is required and the work is therefore accomplished more expeditiously? Or have they ascertained that some inconvenience arises from a frequently repeated impact against the axis of the escape-wheel? As we have experienced some difficulty in attempting to explain this question, we will content ourselves with drawing attention to the fact.

PROPORTIONS IN VOGUE AT DIFFERENT EPOCHS.

684.—TAVAN, JURGENSEN.—Authors and practical men are not perfectly agreed on the subject of the lever escapement, but their differences are not serious; they disagree not so much in what they regard as its principles as in the form, which is varied in accordance with the results of individual experience. *Tavan*, and subsequently *Jurgensen* who merely copied him, prefer the locking surfaces to be arcs of circles concentric with the axis; the pallets thus occasion no recoil, and the unlocking becomes smoother and easier. But we shall in its proper place see that recoil is essential to this escapement.

According to Tavan the driving planes of the pallets should be curved; that of the engaging pallet convex and of the disengaging pallet concave, the radius of curvature being in both cases almost identical with that of the wheel. The rest is tangential, the locking faces being equidistant from the pallet-staff: the energy of the impulse then differs in the two cases.

Jurgensen advocates a curved form for the engaging driving face but makes the other straight. The faces are inclined at an angle of 5° and the same inclination is given to the planes at the extremities of the club-teeth of the escape-wheel. In his pallets the locking faces are not at equal distances from the centre of the staff, occasionally differing by the entire thickness of a pallet. This arrangement is doubtless made with a view to equalize the impulses.

Both authorities fix the total lift at 40°.

685.—MOINET.—The depth of the locking or the pitch of the pallets and escape-wheel should not, according to Moinet, exceed 3° or 4° at most and is often less; the *draw*, or recoil of the teeth during unlocking, from 2° to 3°; inclination of the

front faces of the teeth about 5° (the two latter quantities are not sufficient); and, lastly, the total lift from 50° to 60°.

With the exception of a few practical details that facilitate the planting of the pallets and escape-wheel, the above summarizes nearly all that has been written on this subject. The opinions of practical men conform more or less to what is given above, and, as a rule, each follows a system that is not derived from any theoretical considerations, but which is nevertheless in some cases supported by experience and observation.

CHAPTER II.

RULING PRINCIPLES IN THE CONSTRUCTION OF LEVER ESCAPEMENTS DEDUCED FROM THEORY AND EXPERIENCE.

686.—The main object of this chapter is to set forth and explain the experimental facts that in the workshop are regarded as laws, at the same time extending and correcting them.

We would here repeat, so far as it is applicable, much of the advice already given, and more especially that contained in article **525.**

In referring the reader to these passages, conscious as we are of the difficulty of eradicating prejudices, we would especially insist on an observation which was addressed mainly to designers and manufacturers; it is not by the mere imitation of satisfactory originals but by a thorough knowledge of the laws of motion, by a resort to calculation and an exhaustive study of the theory of escapements, that we can hope to secure the best results; namely, a rate that is very good in the present and reliable in the future.

Opening of the Pallets.

687.—The opening of the pallets, *measured from the middle of the arms* or *from one locking face to the other*, includes two and a half spaces of the wheel; three teeth are then embraced by the pallets. This number is not taken at random; the pallets might embrace a greater or less number of teeth, but in the latter case the pressures would be increased and, the surfaces of

contact being very small, the action would be less certain and mathematical exactness would be essential in the adjustment in order to avoid loss of time; when the number of teeth is greater, the weight of the several parts and the friction would become excessive and we should lose all the advantages that are gained by shortening the escapement arms (**611**).

To fix the centre of the pallets.

688.—The following method is usually adopted.

Assume *f* M U N *r* (fig. 4, plate VII.) to indicate the circumference of the wheel. On this arc mark the two points M, N; the first corresponding to the extremity of a tooth and the second to the middle of the third space from M.

Draw the two radii, O M, O N, and at their extremities the perpendiculars M A, N A. The point of intersection, A, of these two lines indicates the position of the centre of the pallet-staff that would set the escapement tangential. It is only necessary to bear in mind the principles discussed in the Introduction (**73** and **74**) to see that the points M and N, at which the locking occurs, are thus in the most favourable position, regarded solely from the point of view of facility of unlocking.

Is this condition, however, of tangential locking essential, for during the entire locking the pallet is detached from the balance, and therefore the amount of oblique or perpendicular pressure of the tooth is assumed to have no influence, or at any rate very little, on the rate of the watch?

Before answering this inquiry we must consider the two other functions of the pallets namely the *impulse* and the *unlocking;* functions which are usually considered to take place under the best conditions at the same point as the locking.

Should the Escapement be set tangential?

689.—It follows from the preceding considerations that three distinct effects of the escapement should occur on the tangent to the circumference of the wheel or approximately so.

The *locking*, when the tooth of the wheel is held at rest against the pallet;

The *unlocking*, which takes place when the lever, being driven by the balance, causes the pallet-arm to move while its locking face is pressed by the tooth, and thus to bring this tooth to the edge of the impulse plane;

The *impulse*, occurring immediately afterwards, during which the tooth passes along the impulse plane, exerting a pressure against it.

If the necessity of draw be admitted, tangential rest at once becomes impossible, for lockings of this description must occur after the line O I and before the line O J (fig. 4, plate VII.).

From this impossibility it immediately follows that only one point of the surface traversed by the tooth of the wheel during unlocking can be tangential. This point is M on the arm P M, and N on V N (fig. 4).

Finally, it is still open to us to place the middle of each driving plane, M X and N Z, at the point of intersection of the radius and tangent, and then the locking faces S M and V N will be displaced towards the left of M and N, the two points at which the tangents touch the circumference of the wheel.

This simple explanation will suffice to show how erroneous, from a theoretical point of view, is the so-called rule that the escapement should be tangential. It can but enable us to fix some one given point.

What is it that we wish to secure in horological mechanism? Clearly that the uniformity of its indications may continue for a considerable period. But we know that this uniformity is closely dependent on the changes that occur in the ratio existing between the motive force and the momentum of the balance; and we further know that in the course of time the first of these quantities diminishes much more rapidly than the second.

Hence it is of the highest importance to adjust and render as uniform as possible the force by which the impulse is given to the moderator, since it is this that is most rapidly modified by the thickening of oil.

This being clear, we will proceed to enumerate the advantages and objections to making (1) the corners of the locking-faces and (2) the middle of the impulse planes tangential.

Tangential Unlockings.

690.—Let H (fig. 44) be the centre of motion of the pallets; A and B the two points at which the teeth are released from the locking faces; F and O the points of locking; A C the engaging impulse plane, and B D the disengaging impulse plane.

A glance at the figure will show that:

(1) The unlocking takes place on the engaging pallet with

the resistance arm gradually increasing from H F to H A; whereas the second unlocking commences with the longer lever, H O, and terminates with H B that is equal to H A.

Hence the unlocking from the face B O offers the greater resistance.

(2) The tooth acts on the engaging impulse plane at the extremity of a lever which at first measures H A and gradually diminishes to H C: the friction also is that known as engaging. The disengaging impulse plane commences with a lever H B equal to H A, but this gradually becomes longer and the friction is disengaging. The impulse therefore is the more energetic on this latter plane.

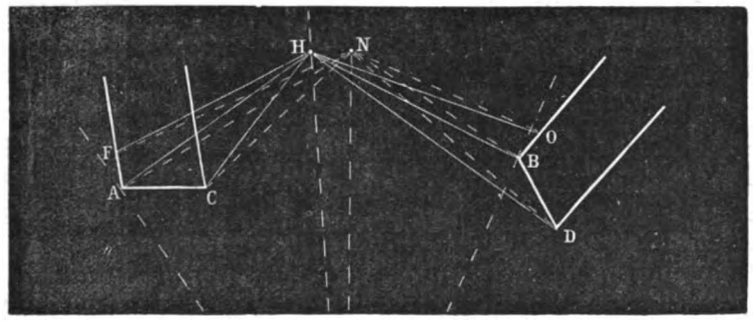

Fig. 44.

Hence we see that: *the impulse of least energy is preceded by the unlocking of smallest resistance and vice versâ.*

Tangential Impulse.

691.—If it be required that the middle of each impulse plane be tangential, the centre of movement of the pallet should be transferred from H to N (fig. 44).

As in the preceding case, a simple inspection of the figure will show that:

(1) The lever arms N F, N A, are much longer than N O, N B, and therefore the unlocking from the engaging pallet will offer a resistance considerably in excess of that opposed by the disengaging pallet;

(2) Since A N is equal to N D and C N to N B, it follows that the application of the impulse commences under rather more favourable conditions with the plane A C than with B D; but the friction on A C is engaging, so that (providing there is no slipping action nor loss of time) we may regard the two impulses as practically of equal value.

Thus with this second arrangement: *the same amount of impelling force is called upon to overcome, in the two cases, resistances that vary considerably.*

Ignoring for the present the influence which this inequality may have on the action of the balance-spring, we would merely remark that the motive force falls off in time much more rapidly than the force that effects the unlocking; the proportion therefore initially existing between the motive power and the energy of the balance will be modified sooner and more seriously and this second arrangement will render necessary greater care in the selection of the balance-spring.

Conclusion. The practice of making the edge of the locking faces tangential will be seen to possess advantages, but it is essential that the draw in the two cases secures equal steadiness of the pallets. Makers experience some difficulty, however, in adjusting the amount of draw since one arm is rather longer than the other, and they prefer the arrangement in which the middle points of the impulse planes are tangential. Nevertheless the system with unequal arms has been generally thought best.

We venture to trust that every watchmaker will have clear ideas on this much vexed question after reading the following articles.

Errors in construction may cause the unlocking of least resistance to occur after the weaker impulse.

692.—In our first edition we stated, on the authority of several of our best horologists, that "experience is, as a rule, in favour of equal impulses," in other words, of the arrangement in which the middle points of the impulse planes are tangential. Careful observations have since removed any doubts that we had on the subject, and an exhaustive study of it has led to these two facts: (1) the tangential position of the edges of the locking faces has advantages; (2) the conclusion we first arrived at was too positive, for we did not fully appreciate the objections to serious inequalities in the resistances.

At the same time the importance of this inequality must not be over-rated, for with actions that are so instantaneous it is impossible to observe with certainty the regularity or irregularity in the movements compared. As we have observed, watches are met with where the pallets are formed in accordance with the principle of equal impulses that possess excellent rates.

But on minutely examining the action of a number of these escapements we have observed in several either a difference in the draw on the two pallets or a loss of force on one of the planes by slipping, etc., which had the effect of causing the unlocking of least resistance to occur after the weaker impulse; a circumstance that was none the less efficient through being accidental.

We trust that our explanations of this vexed question of the comparative advantages possessed by the two systems are sufficiently exhaustive to enable every intelligent watchmaker to form clear ideas in the future regarding the entire question, and this should be especially the case after he has read the articles on draw; we will here make a practical suggestion equally applicable to other escapements.

Practical Test of the Energy of the Two Impulses.

693.—When the mainspring is fully wound up, mark on the plate of the watch the extreme point reached by a dot on the balance in the longer of its two half-vibrations. After removing the pallets bring the mark on the balance above that just made on the plate, then release it and make a second mark on the plate at the terminal point of the vibration.

These two points will be useful for testing the equality of the two impulses as well as for indicating which requires adjustment. The action of the balance-spring will be all the less constrained according as the balance when engaged with the escapement approximates towards the condition of a detached balance.

To determine the Draw or Recoil.

694.—Formerly the locking faces of the pallets were circular arcs concentric with the pivots (*a m v*, *c n v'*, fig. 2, plate VII.); the wheel did not recoil at all during the act of unlocking and this was, in consequence, accomplished with very great ease. But this facility was itself a fault that had to be avoided by so forming the locking faces as to cause an appreciable recoil of the wheel during the unlocking. The unlocking with recoil involves the employment of a rather greater motive force than was formerly employed, and this evil cannot be avoided; for, unless the amount of *draw* be made considerable, the pressure of the tooth against the face of the pallet is not sufficient to maintain the lever against the banking; so that when subjected to a shake it moves away from the banking, and the guard-pin or

dart comes in contact with the roller edge. If such an accident occur frequently it becomes impossible to time the watch.

695.—With a view to obtain the requisite amount of draw, watchmakers made pallets with their arms straight instead of curved as was formerly the practice, the locking faces being inclined so that the locking point was as far within the circular face as it was desired the wheel should recoil (that is, measuring the interval between the straight locking faces and the curves *a m v*, *c n v'*, fig. 2).

Experience alone can indicate what is the most convenient amount of recoil.

In practice 12° was allowed on each arm. Such an amount would appear to justify the hope that the two draws would be equal, but this is not secured when the pitch of the teeth and pallets is shallow.

The method ordinarily adopted for making pallets consists in setting the cutter so that it forms an angle of 12° with the radius of the wheel prolonged (O M I, O N J, fig. 4, plate VII.). The two planes B M, D N, are thus obtained, representing the locking faces of ordinary pallets in which a recoil of 12° is produced with each arm. A careful examination of the drawing, avoiding confusion amongst the several lines that meet in a point, will show that the draw on the right-hand arm is very different in character to that on the left-hand, and that these two effects occur with levers of unequal lengths.

696.—The following demonstration will be found to be conclusive if attentively considered.

From the centre A draw the circular arcs C M P, E N H, through the points M and N (in the same figure), and from the definition of a tangent (**72**) it follows that the prolonged radii O M I, O N J will only touch these arcs in the points M and N at which they are cut by the lines A K, A L; hence, beyond these points, the arcs diverge more and more from the prolonged radii M I, N J. This being granted, and observing that the extremity of each arm during the motion of the pallets traces out one of the circular arcs just referred to, it is at once evident that the wheel will recoil through the entire space included between the locking face N D and the arc N H; but since this arc, starting at the point N, always diverges from the line N J which is inclined at an angle of 12° to the locking face, it is perfectly manifest that the recoil must exceed 12° by an amount which increases as the extent of motion

of the pallets becomes greater. Under ordinary circumstances this recoil may be assumed without sensible error to actually amount to 15°.

Examining now the action that occurs on the left-hand locking face, we see that the wheel will recoil through the interval enclosed between the plane M B and the circular arc M P, and that this arc always falls short of the line M I except at the point M; the recoil, therefore, must be less than 12°. This difference may be estimated at 2° or even 3°.

These facts afford an explanation of the observation familiar to all watch-jobbers; namely that, in nearly all the older lever escapements produced in Swiss factories, the draw, while being excessive on the exit pallet, is almost nothing on the other. We believe we were the first to explain in what lay the error on the part of the makers and at the same time to indicate the correction to be applied.

In order to effect an equilibrium the usual practice at the present day is to increase the inclination of the left-hand face to 15°, that is to say, the angle I M B is 15°.

697.—The above demonstration is exact when the pitch of the pallet and escape-wheel is somewhat deep, the locking faces being straight; but with the modern practice of bringing the locking points very near to the corner of the pallet, so that at those points the curves P M, H N (fig. 4), and the straight lines I M, J N, practically coincide, this difference of three degrees in favour of the engaging pallet must rather be corrected by making the arms A M, A N, unequal.

In conclusion, the practice adopted in the factories is a mere experimental deduction which is sufficient for ordinary watches; but the maker should be able to determine in anticipation, by means of a large-scale drawing made in accordance with the principles of the preceding theory, the conditions that secure equal resistance to unlocking on the two arms and maintain the steadiness of the lever against the bankings invariable (**630** and **635**).

This suggestion is all the more pertinent here, because we shall presently see that the inequality in the two draws is not entirely due to the source of error above indicated; it very often arises from faulty construction, especially as regards the opening of the pallets and the pitching of the escapement.

The great amount of draw renders necessary a consider-

able motive force, and consequently the watch must have some thickness. When these conditions are not fulfilled, the lever escapement never works well.

Change effected in the draw by varying the interval between the centres of motion.

698.—In consequence of the division of labour in factories the pallet-maker is not the workman that pitches the escapement in the watch. Thus it happens that when the pallets are not in perfectly strict proportion, the escapement-maker is compelled to vary the length of the ruby pallets and pitch the mobiles in any position that will ensure the escapement working well. So far from concerning himself about tangential escaping, he utterly ignores it, and indeed the great majority of them could not explain the meaning of the expression.

This, however, would occasion no serious inconvenience, since such a tangential escapement as has hitherto been advocated in the workshop is quite a delusion (**689**), were it not that a very objectionable fault results from it.

The maker forms his pallets either according to a rule that sets the middle of the incline tangential, or to one that requires the edge of the locking face to be so, and the centres of the wheel and pallets are therefore fixed.

It follows that if the escapement-maker does not set the centres of motion at the distance apart previously determined upon by the pallet-maker, the draw on the two faces is not such as it was intended to be, and the steadiness of the lever is less secured on one side than on the other. It may even happen that on one side this steadiness does not in practice exist.

DEMONSTRATION.

699.—Assume *r a* and *n s* (fig. 5, plate VII.) to be two locking faces making the angles *z v a* of 43° and *x t n* of 40° with the radii L *z*, L *x* that pass through the tangential points. This escapement is adjusted for a distance between centres equal to C L.

The tooth will recoil by the two amounts *a z* and *n x*.

Examine the manner of resolution of the force acting on either plane. Taking the lines *a o*, *n i*, to represent this force in magnitude and direction, the parallelograms *a b o d*, *n* C *i j*, will indicate the manner in which it is resolved (**69** and **228**).

The drawing shows that the line *b o*, as compared with *b a*, may be taken to indicate the portion of the force that presses

the lever against the banking q; and that the ratio of C i to C n gives a measure of the force pressing the lever against the pin q'.

The force then acting on q' through the pallet $t\ s$ is rather less, but since it is applied at the extremity of a longer lever, we may admit that the two draws secure very nearly the same degree of steadiness of the lever.

Assume that the opening of the pallets, measured between the two locking points, through some error of construction, slightly exceeds the amount $a\ n$ given in the figure, a circumstance which very often occurs in practice. The escapement-maker will simply replace the two ruby pallets by longer ones and will move the centre of the pallet-staff from C to, say, K.

This displacement, apparently so natural and simple, will nevertheless entirely disarrange some of the essential functions of the escapement.

The direction of the impelling force (represented by the equal lines $a\ o\ g$, $n\ i\ p$) will be unaltered; we therefore know from the parallelograms $a\ f\ g\ h$, n K $p\ m$ how this motive force is distributed on the two pallets.

An examination of the several lines in the figure compels us to recognize the fact, utterly unexpected by the majority of watchmakers, that the force maintaining the lever against the banking q is enormously increased, whereas that which should hold it against q' has not only diminished but actually tends *in an opposite direction.*

The direction of the motive force is *above* the first axis C but *below* the new axis K and the steadiness of the lever, when the tooth acts at n, is so indifferently maintained that a slight shake would in very many cases suffice to occasion contact between the guard-pin or dart and roller.

Remark.—The reader should attentively consider the above demonstration: he will see that the recoil of the wheel cannot be regarded as a strictly accurate test of the steadiness of the lever against the banking. For the recoil with the arm $t\ s$ is considerable, since it is represented by the interval between the circular arc $t\ u$ drawn from the centre K and the straight line ts, and yet the effect which the draw is intended to produce is, notwithstanding this great amount of recoil, either zero or negative.

The Lifting Angle.—Experimental Data.

700.—The extent of lift is dependent on various circumstances which have been already enumerated in **96** and the suc-

ceeding paragraphs, and yet it has long been accepted as a rule that 40° should be invariably employed. If makers had observed that in the lever escapement (1) the escape-wheel transmits the motive force to the balance through an intermediate piece by which a portion is absorbed; (2) this intermediate piece, owing to its size and weight, has a certain amount of inertia, that is to say it opposes an appreciable resistance to any sudden change in its movement; and (3) that the balance travels with a considerable velocity because, while having the same number of vibrations as other escapements, their amplitude is greater; if, we say, they had observed these facts, they would have seen that, when the impulse commences, the lever acts with very little energy on the balance, especially if the inclination of the impulse plane is slight, and that it is only really effective towards the end of the lift. Hence it follows that a lifting angle of 20° on each arm, or a total of 40°, will not suffice for the majority of lever escapements that are made.

701.—The mean value of the lift may be as much as 50°; rather more (55°) for small escapements and somewhat less in the case of large ones (45°).

It is easy to explain the cause of the difference observed in the above numbers in accordance with what has been already said when discussing the cylinder and duplex escapements, remembering that, since small balances have less inertia than large ones, they require an impulse of longer duration and more frequently repeated; on the other hand, with large balances, possessing some mass and a comparatively greater power of maintaining the velocity they have acquired, the extent of lift must not exceed what is absolutely necessary (**323**, etc).

Remark.—But, we would again observe, the figures given cannot be looked upon as in any way absolute. As a general rule, when the escapement is accurately constructed and the mobiles light and in accordance with the requirements of theory, an arc of moderate extent (from 35° to 45°) is all that is required. On the other hand this lifting angle must be increased (from 45° to 55° or even 60°) in heavily made lever escapements and in those in which a considerable proportion of the impulse of the wheel is lost owing to its being too much decomposed through faults in construction or arrangement (**97** and **98**).

To ascertain the height of the Impulse Plane and the position of the Ruby-Pin.

702.—Tavan and Jurgensen give 40° as the limit of the total lifting arc, whereas Moinet allows between 50° and 60°; we have already explained this difference.

The lifting angle, as measured by the extent of the arc traversed by the balance between two unlockings, having been determined upon, the inclination of the impulse planes is adjusted so as to secure this amount of angular movement. Varying the distance between the ruby-pin and the centre of the balance will also modify the lifting angle. The inclination of the impulse planes and the position of the ruby-pin must then be correlated.

Height of Inclines.

703.—The lever will be driven more and more backwards as the impulse plane is more elevated; it is, then, the height of this incline that determines the extent of the angular movement of the fork. The force opposing unlocking and the resistance at the end of the lift increase as this movement becomes greater, and, on the other hand, if the path of the fork is very short there is much risk of the lockings and of the entrance of the ruby-pin within the notch not being performed with sufficient certainty. Moreover, if we consider an escapement beating 18,000 vibrations per hour, it is observable that the tooth traverses an impulse plane of slight inclination with great rapidity and that it only attains a considerable momentum when about to fall against the locking face; the blow thus occasioned is detrimental for it gives rise to quivering of the pallets and may cause the guard-pin or dart to come in contact with the roller.

In the first case (with an excessive inclination of the impulse plane) the watch sets as soon as the oil becomes at all thick; in the second case (with the incline too low) the watch will be a very bad timekeeper and the amplitude of the vibrations insufficient (**247** and following articles).

Experience can alone decide on the best intermediate point between these two extremes, and it has proved that, as a rule, we must impart a total angular movement of 10° to the pallets and lever; that is to say, each impulse plane should be such that, when it is impelled backwards by the tooth, the fork is caused to traverse an arc of 5° on either side of the line of centres.

704.—The following method is usually given for determining the inclination of these planes.

From the centre of the pallet-staff A, fig. 4, plate VII., draw the two lines A X, A Z, inclined at angles of 5° to the tangents A M K, A N L; join M X and N Z, and the planes M X, N Z, thus determined will be such as to cause a movement of 5° with each arm.

This method is fallacious and the results obtained are inexact. Not only are the two half-lifts inferior to 5°, they are not even equal to one another.

For the line A M K is a tangent to the circumference of the wheel and therefore only touches it in one point M (**72**). Hence the tooth can merely repel the impulse plane up to this circumference and towards the end of the lift the entire space included between the arc M U and the line K M A is lost, a loss which may be approximately estimated at one degree.

The same line of reasoning will show that the lift on the exit pallet is very similar. Hence the total movement, nominally 10°, is in reality not more than 8° or 8·5°.

It is surprising that this difference, which is very conspicuous when the locking points of the escapement are set tangentially, has not been previously observed. But this fact may be explained by the following circumstance; the methods adopted in practice, as we shall subsequently see, involve the testing of the inclination of each impulse plane separately and thus eliminate the error involved in the above theoretical method. It is, however, very easy to represent the exact proportion in a drawing by proceeding as follows.

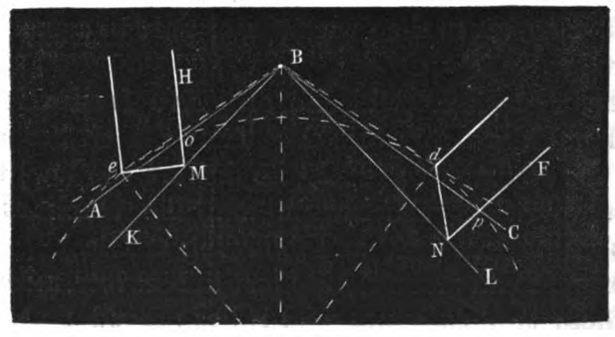

FIG. 45.

From the centre (B, fig. 45) of the pallets draw the two lines B A, B C (which must not be confounded with the tangents B *e*, B *d*) passing through the points *o* and *p* where the faces M H,

N F of the pallets cut the circumference of the wheel. The two lines B K, B L, must now be drawn inclined at angles of 5° to these lines B *o* A, B *p* C. By joining *e* M and *d* N we obtain the required impulse planes. A simple inspection of the figure will show that when the wheel advances the engaging pallet will move through the angular space *o* B M, and the disengaging pallet through *p* B N. The two lifting angles are therefore equal, and the angular movement of the lever will actually be 10°.

Apparent and Real Lift.

705.—If the corners M and N always lie on the lines B M K, B N L (fig. 45), the movement of the lever will be 10° at whatever points the impulse planes commence; they may start from points *o*, *o'*, on the circumference of the wheel (fig. 46), beyond it at *v*, *v'*, or within it at *s*, *s'*. But then, as in the case of the cylinder escapement (**305**), we must distinguish between *real* and *apparent lift*.

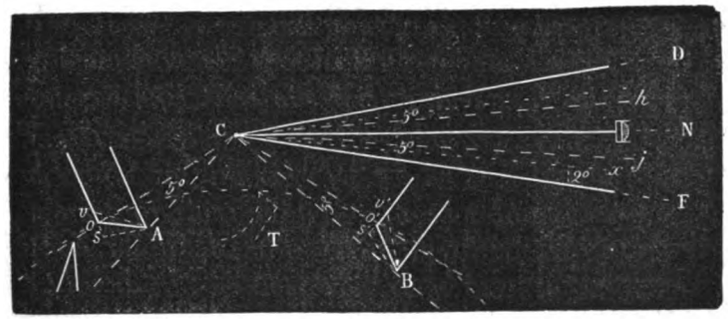

Fig. 46.

Consider the action of a *pointed* or ratchet-tooth wheel.

If the impulse planes spring from the circumference of this wheel, at *o* and *o'* (fig. 46), each half-lift will commence with the middle of the lever placed accurately on the line of centres C N. The half-lift, then, is exactly 5°.

When they commence 2° beyond the circumference, at *v* and *v'*, the half-lift will commence when the lever is still about 2° short of the line of centres; the actual half-lift is, then, 7°, or *h* C F.

If, on the other hand, the planes only commence 2° within this circumference (at *s* and *s'*), the half-lift will be delayed until the middle of the lever has got 2° beyond the line of centres; and therefore, since the movement of the lever is 5° from the line C N, the half-lift will in fact be 3° or *j* C F.

706.—We know that the energy of impulse bears a definite ratio to the height of the impulse plane (**247**). Bearing this in mind, let us consider three escapements absolutely identical except that the first has driving planes *s* A, *s'* B (fig. 46); the second *o* A and *o'* B; and the third *v* A and *v'* B; we see that the *apparent* TOTAL LIFT is, for all three cases, 10°, whereas the *real* TOTAL LIFT, representing the heights of impulse planes, will differ in the proportion 6 to 10 to 14. They will require unequal motive forces.

707.—If, instead of being pointed, the teeth are clubbed, terminating in a short inclined plane of 2° inclination as represented at T, fig. 46, the only difference will be that the half-lift commences 2° before the centre with the planes *o* A, *o'* B, 4° before with the planes *v* A, *v'* B, and exactly at the centre with the planes *s* A, *s'* B.

This fact gives an easy means of practically ascertaining whether the height of the impulse planes is correctly adjusted.

708.—It is doubtless unnecessary to observe that the entire amount *o s* and *o' s'* (fig. 46) by which the impulse plane pitches within the circumference of the wheel when the inclinations *s* A, *s'* B, are employed, will be added to the periods of locking of the escapement, and on the other hand these intervals will be diminished by *v o*, *v' o'*, representing the extent to which the impulse planes reach beyond the circumference of the wheel, if *v* A, *v'* B are the forms adopted; but, although the half-lift may vary, the total lift is always 10°.

It will thus be seen that, by varying the inclination of either the pallet or tooth, we are able to change the ratio of the extent of locking to the half-lifting angle.

In an escapement with impulse planes *o* A, *o'* B, each giving a lifting angle of 5°, and in which the club-tooth of the escape-wheel has an inclination of 2°, the motion of the lever will be divided as follows:

The locking will occur when the face *v'* is in the position *s'*, and the balance, on returning so as to accomplish the unlocking, will drive the lever to C *x*. The fork will next act on the ruby-pin from C *x* until the line C D is reached, in other words, through an angle of very nearly 8°. Since the total angular movement of the lever is 10°, we see that about *one-fifth* will be devoted to the unlocking and *four-fifths* to the impulse.

This example will be sufficient to show how important it is to distinguish between the apparent and real lift.

On curved and straight impulse faces.

709.—Tavan, as we have already seen (**684**) preferred that the impulse faces should have a curved form; that of the engaging pallet being convex and of the disengaging pallet concave, while the radius of curvature of both was that of the wheel itself. Jurgensen made the first of these curved and the second straight. The majority of watchmakers make both faces straight.

710.—There is no doubt that by giving them a certain curvature it is possible to obtain regularity in the lift, that is to say so to adjust the velocity of the wheel that it always bears a certain pre-determined ratio to that of the balance; in other words, the movement of the escape-wheel may be accelerated, uniform, or retarded as compared with that of the balance.

But here we are met by the insuperable difficulty of forming such minute curves accurately, and their theoretical advantages would always be nullified by faults of construction.

711.—Moreover, with clubbed teeth, the lifting action on each pallet may itself be regarded as consisting of two portions.

In the *first*, the front corner of the tooth traverses the entire length of the impulse plane. In the *second*, the small incline of the tooth slides past the corner of the pallet. During these two portions of the lift equal spaces are traversed in the majority of modern escapements, for the length of the incline of the tooth is usually made the same as that of the pallet. Hence the curved pallets could only regulate about the first half of each half-lift.

712.—*With a curved impulse face* on the engaging pallet, the wheel will travel slower at the commencement of the lift (as compared with a straight face) and more rapidly towards the end according as the curvature is more pronounced. The impulse will then possess less energy, and the drop will be the more severe.

Assuming the possibility of giving the required curvature to the face, the only advantage it would possess would be to very slightly facilitate the starting of the escapement; but this unimportant advantage would nearly always be accompanied by a more severe drop and a loss of impelling force.

713.—*With a concave impulse face* on the disengaging pallet,

the wheel would travel somewhat more rapidly at the commencement and slower toward the termination of the *first portion* of the lift than if the incline were straight. The *second portion* would be accomplished in the two cases in very nearly the same manner.

It appears then that a slight advantage might be secured by employing a concave face, but it is to be observed that the curvature would have to be so little that there would be very great danger of making it excessive and the starting of the escapement would then be rendered more difficult.

714.—Curved faces have important advantages when their extent is such that accurate workmanship is possible; but when an action so short, as regards both time and space traversed, as the lift of the pallet in a lever watch is under consideration, what can be looked for?

715.—It is the safest plan to make the impulse faces of the pallets straight, for on the whole their construction is easier and they offer advantages at any rate equivalent to those anticipated from the use of curved inclines.

After the completion of these straight faces they should be rounded off crosswise in order to diminish the extent of the contact of the tooth and pallet. We shall again refer to this beading.

To fix the position of the ruby-pin and the length of lever necessary to secure a given lift.

716.—The angular movement of the lever being, as we have already seen, fixed at 10°, the following is the method usually recommended for fixing the position of the ruby-pin or its distance from the centre of the balance-staff, the three centres of movement being supposed fixed.

From the centre B (fig. 2, plate VII.) of the pallet-staff draw the line B X, inclined to the line of centres B P at an angle of 5°, that is to say the angle through which the lever moves on one side of this line.

From P, the centre of the balance-staff, draw P Z inclined to the line of centres P B at an angle equal to half the total lifting angle; thus, with a total lift of 40°, the angle Z P F will measure 20°.

The point I, where the lines B X, P Z cross, gives the position of the ruby-pin, that is to say the distance between its *point of contact* and the centre of the balance.

If, instead of 40°, the lift were required to be 45°, 50°, 60°,

the angle Z P F would be 22·5°, 25°, or 30°, the ruby-pin being thus brought gradually nearer to the centre of the balance.

We have italicized the words *point of contact* because the centre of the pin will not be on the circumference described by the radius P I except in the case of a round pin, and at the present day it is always made oval or triangular.

A careful examination of figs. 1, 2 and 3 of plate VIII. will make evident the functions and mode of action of this ruby-pin.

Length of the Lever.

717.—The method adopted for determining the position of the ruby-pin also gives the distance between the centre of the pallets and the fork.

It is only necessary to describe an arc Q I T (fig. 2, plate VII.) from the centre of the pallets and with a radius B I in order to ascertain this length, measuring from the centre to the point at which the prongs commence.

718.—The following practical rule is usually employed for ascertaining this length. It is deducible from the above reasoning.

For a lift of 40° divide the centre distance B P (fig. 2) into *five* parts and deduct one of these portions from the end P.

For a lift of 45°, divide into *eleven* parts and deduct two. (Or divide into five and a half parts and take four and a half.)

With a lift of 50°, *six* parts and deduct one.

With a lift of 60°, *seven* parts and deduct one.

These distances can be easily measured either by using a finely graduated brass rule or, preferably, a douzième gauge the smaller arms of which are pointed in order to serve as a compass.

The diameter and form of the ruby-pin and the width of the notch in the fork.

719.—Article **716** gives a graphical method of ascertaining the position of the ruby-pin and it only remains to determine its form and diameter.

The *unlocking* would take place under the most favourable possible conditions if the side of the notch were struck perpendicularly by the ruby-pin, when over the line of centres, B P, the friction being disengaging (**73**).

In order that the *impulse* may be tangential and the friction disengaging, the back of the pin should, at the end of the unlocking, be on the line of centres, in which position it would receive the perpendicular blow from the other side of the notch.

It will easily be seen that these two conditions could not be satisfied, since the movement due to the impulse and the unlocking extends over a space greater than the actual width of the ruby-pin.

The energy absorbed and decomposed during the *unlocking* becomes greater as the contact occurs farther from the line of centres, and the action is the more oblique to this line. Hence the resistance opposed to the ruby-pin during the unlocking becomes less as the angular movement of the lever is reduced. It thus becomes important to avoid increasing this movement beyond the 10° which are known to be necessary, and not to set the bankings (K, L, fig. 2, plate VII.) too far apart as this would needlessly increase the resistance opposed to the pin in its engagement with the notch.

720.—An examination of fig. 5, plate VI., shows that as we increase the width of the notch the side *s n* will gradually approach the line of centres P R, and the engaging friction during unlocking will be reduced in proportion.

721.—We have seen from one of the preceding articles (**705**) that, when the escape-wheel has pointed or ratchet teeth, the impulse commences with the middle of the fork on the line of centres, whereas with club teeth this action commences about 2° beyond it. It results then that, in the first case, the side of the notch is separated from the line of centres by a distance equal to half the width of the ruby-pin and by this amount plus 2° in the second case. A reduction in the size of the pin or, what amounts to the same thing, a reduction in the width of the notch, increases the efficiency of the impulse.

722.—Since a very wide notch facilitates the unlocking and a very narrow notch facilitates the impulse, it becomes evident that the width would have some mean value balancing, as far as possible, the resistances during the two actions, were it not that it is determined by a third condition of primary importance which we proceed to consider.

Size of Ruby-Pin.—Experimental datum.

723.—If we assume the lever to rest against the banking L (fig. 5, plate VI.) and make the notch at first *e a* and subsequently *m n*, which is three times *e a*, we shall find that in the first case the thickness of the pin is rather less than the interval included between the circular arcs B and C, whereas in the

second case it would nearly equal the distance between the arcs A and D. With the narrow notch the pin would require to be very thin in order not to catch against the corner *e*, and, since the face *i a* of the notch against which the pin acts would be very materially reduced, the action would become uncertain. With the larger notch *m n*, the surface *s n*, that engages with the ruby-pin, is of such extent that the certainty of the action is guaranteed, even when a considerable amount of freedom is allowed between the pin and the corner *m*. This example proves that the performance of the fork and roller is rendered more certain as we increase the width of the notch, the diameter of the ruby-pin becoming at the same time proportionately greater.

Experience can alone determine what is the most convenient diameter of the pin; and we have just seen (**721**) the disadvantages of one that is too large. It has shown us that, in the escapement of an ordinary watch, where the pallet-arms and teeth are approximately of the same width, the width of the ruby-pin (that is, its greatest diameter since it is either oval or triangular) must be slightly in excess of the breadth of a pallet; in other words, the diameter of the pin is about equal to one-third of the interval between the points of two teeth of the escape-wheel. Every intelligent watchmaker will be able to see when he can advantageously reduce it a little below this amount.

The form of the Ruby-Pin.

724.—It must not be cylindrical. When one is met with of this form it should be flattened at the front and set somewhat forward, or it will be better to replace it by one of suitable form.

A watchmaker employing a round pin will be obliged to increase the angular movement of the lever so as to prevent the superfluous part, H (fig. 6, plate VII.), from coming in contact with the corner of the horn. Any such increase in the extent of path of the lever increases the harshness of the friction on the ruby-pin entering the notch, and, further, if the width of the notch is adjusted for a less path, he will find that towards the end of the lift it is not the fork that drives the ruby-pin, but that this ruby-pin carries the lever along with it; and such an action is very prejudicial.

When the front of the pin is flattened the freedom is much greater than with a round pin, but even this is not the best possible form to give it.

In the first edition of this work we recommended a

flattened oval section, the flattening being more marked on the front face in order that only a small extent of a rounded surface might come into action.

At that time the triangular pins were very frequently cut from cylinders; and, as a consequence, the side of the notch was, in numerous instances, struck by a nearly flat surface to effect the unlocking, so that this surface was liable to adhere to the fork. Moreover at the beginning of the lift the fork itself struck the pin on an angle.

At the present day it is the usual practice to cut the triangle from an oval shaped pin, as by doing so both the above inconveniences are avoided.

The watch manufacturer will find it useful to draw the fork and ruby-pin on a large scale in their successive positions.

A notch that becomes wider towards the base or is dovetailed is found to reduce the play of the ruby-pin, but it requires very perfect workmanship.

On the distance between the centres of the Escape-wheel and Balance.—Long and Short Levers.

725.—It seems at first sight to be a matter of indifference whether the centre of the balance is set near to or far from the centre of the pallets; in other words, whether the lever be long or short, since the distance of the ruby-pin from the balance-staff will vary directly with the length of the lever, and thus the relative lengths of the two levers will remain unchanged; hence the power and the resistance are always in the same proportion if the lengths of the levers are only considered, and it would therefore appear that the results will be identical.

But it must be remembered that, when the lever is long, (1) the whole will be heavier, not merely in consequence of the additional metal required to lengthen the lever, but also through the balance-weight at the outer end which must be added in order to maintain the equipoise; the inertia of the mobile, and therefore its resistance to being set in motion, will thus be considerably augmented; (2) friction occurs over a greater extent of surface; (3) increasing the length of fork involves the use of a larger roller. This large roller, by needlessly increasing the weight of metal at the centre of the balance, compels us to increase the rim of the balance in proportion (**45**), and to use a stronger balance-spring. In short, it becomes necessary to apply a greater motive force, since all the resist-

ance throughout the escapement is increased and the sources of irregularity will be proportionately more marked.

These considerations show that an advantage is secured by keeping the length within moderate limits, but it is important to avoid falling into excess in the opposite direction. With a very short lever the acting surfaces are very much reduced; all the parts therefore must be made with the greatest accuracy, the several freedoms become much less, and the regularity of movement is uncertain. We would further add that, in certain callipers of watches, there is some difficulty in pitching the balance since it comes too near to the axis of the escape-wheel; but the principal objections to short levers consist in the *loss of time* for the performance of its functions, and the *excessive pressures*, that ultimately occasion irregularities of timing.

726.—An empirical rule requires that the distance between the centres of balance and pallet-staff should never exceed the *diameter of the escape-wheel* or, at most, this diameter plus one-twelfth.

A second rule, also deduced from observations on numerous watches, limits the length of lever to *the radius of the balance minus the centre-distance of the ruby-pin.*

These proportions permit of considerable latitude, for, in the watches generally produced of the same external dimensions, large and small escape-wheels are met with in conjunction with balances of varied diameters; they give, however, a mean size that is quite sufficient for the demands of daily work.

727.—A manufacturer of high-class watches desiring to construct a standard escapement may accept these rules as starting points, but this first approximation must be subsequently modified

(1) In accordance with the results obtained in reference to the resistance of the acting surfaces, the degree of accuracy attainable, and the insensibility of the arrangement to variations in the motive force. This force must be tried both when the mainspring is fully wound up and when the arcs of vibration are reduced, owing to thickening of oil, after the watch has been going for a period of average duration.

(2) He must, moreover, carefully study a watch of similar dimensions which, after going for a long period, does not exhibit any signs of wear at the surfaces of contact. When the motive force of the watch under construction is inferior to that of the watch used for comparison, the maker should reduce the length

of the new lever in proportion to the diminished force, and conversely in the opposite case. In either case he will, by this means, prevent the pressure exerted against the sides of the notch (which gives rise to the most detrimental friction that occurs in the entire escapement) from exceeding a given amount, already tested and found to be proportional to the resistance offered by these sides. Regularity in the action of the ruby-pin will thus be ensured.

An escapement can be timed with equal facility whether its lever be long or short providing all the parts are well finished, but superiority, that is to say a minimum of variation in the rate and in the resistance of the parts to wear, is always found where the total friction, the pressure and the loss of time are least.

Reasons for the inconveniences arising from the use of short levers.

728.—The earlier lever escapements of Robin's design were provided with very short levers. A. Breguet, who unquestionably perceived their inconvenience, prolonged this portion of the mechanism, and the Swiss makers, carefully following in the track of that celebrated horologist, brought the long lever into favour; in a few years, however, this increase was carried to an excess.

When the discussion began amongst watchmakers as to the advantages that result from reducing the length of the pallet-arms in the Graham escapement for clocks, it was but natural that these novel ideas should be applied to the lever in watches; and, relying on an erroneous theory, they went to an opposite extreme. At the present day numerous makers have, without any intelligent reason, adopted extravagantly short levers.

We proceed to explain the objections to their use; but we feel compelled to repeat what unfortunately requires constant repetition: neither in theory nor practice is there such a thing as a long or a short lever. There is one length that secures the greatest regularity, namely the one that satisfies the conditions laid down in article **727**.

729.—Let D, (fig. 47) be the centre of the pallet-staff; H and N two successive positions of the staff of one and the same balance. The distance D N, is double of D H. Since the angle *l* D N represents the angular movement of the lever in the two cases, we get the proportion:

$$D\,j : D\,l :: j\,H : l\,N,$$

and from this it appears that, as regards the relative lengths of the arms, it is a matter of indifference whether the balance is at H or N.

It has been already pointed out that a long lever increases the resistance due to inertia and the extent of acting surfaces; this then does not require further demonstration (**725**).

Now consider the short lever.

The impulse arm H *j* is half N *l*. Since they are both carried by the same balance, the pressure exerted by the shorter arm will be double that with the longer, and the friction at *j* will be very much more intense than that at *l* (**38**).

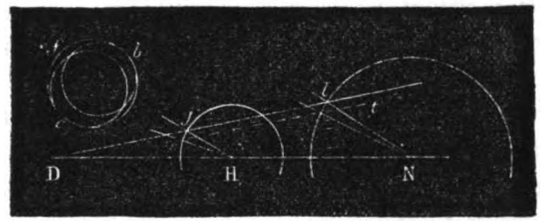

Fig. 47.

For a given amount of shake of the pivots in their holes, with the short lever it will be necessary, providing all the parts are made in proportion, to allow much more freedom between the acting parts. For represent one of the pivot holes by *f c b*. The two inner circles indicate the successive positions occupied by the pivot before and after unlocking begins. By a change of the point of contact of the pivot from *b* to *c* each centre will be displaced by an amount that is relatively twice as great in the case of the shorter arms.

Lastly, if the interval between the lines *j l*, *s t*, be taken to indicate the freedom necessary between the ruby-pin and the notch of the lever, an inspection of the figure will suffice to show that about one-sixth of the work will be wasted with the long arm and very nearly one-third with the shorter. The loss of time will be considerably greater when this latter is employed.

730.—It thus appears that, without taking faults of construction into account, although they are generally numerous with short levers, the employment of these latter involves a more rapid wear of the acting surfaces and renders very slight shake and freedom imperatively necessary not only at the pivots but also between the ruby-pin and the notch of the fork; and this causes the escapement to be far more sensitive to the thickening of oil.

The force supposed to be gained is in reality converted into excessive pressure and impacts.

Size of the Table-roller.

731.—The depth of the notch in the fork should be no more than is necessary for the free passage of the ruby-pin; for when the notch is cut deep it increases the interval between the guard-pin (or dart) and balance-staff, thus rendering a larger roller necessary. The most convenient size for this latter is determined as follows:

The line C J (fig. 4, plate VIII.) indicates the extent of angular movement of the lever (5°) on one side of the line of centres; from the centre of the pallet-staff draw the circular arc *h d* passing through the point of the guard-pin or dart C. The point of intersection of *h d* and the line C J gives the radius A C of the roller, having its centre at A.

The width of the notch in this roller (termed the *passing-hollow*) should be about one and a half or two times the diameter of the ruby-pin. We say *about* because it will increase or decrease according to the extent of angular movement of the lever and the diameter of the roller. It will be subsequently shown how the dimensions of this passing-hollow can be practically determined.

Size of the Safety-roller.

732.—It is usually the practice to make the radius of the safety roller two-thirds of the centre-distance of the ruby-pin.

Thus, when this pin lies on the circular arc T T′ (fig. 1, plate VIII.), that is at a centre-distance B T′, the size of the roller is given by the circle described with a radius B N, which is equal to two-thirds B T′. The intersection of this circle with one of the lines C M or C N gives a means of describing the arc M *r* N and this indicates the distance between the centre of the pallet-staff and the point of the guard-pin or index piece.

The form of the teeth and their inclination.

733.—The teeth of the wheel are either pointed, or large and expanded at their extremities as shown in fig. 2, plate VII.

If pointed they should be thin and elongated; when the ends are clubbed the teeth must be cut away towards the base and no more metal left than is required for rigidity (**91**).

734.—The front face of the tooth (A B, fig. 3) should be in-

clined forward, so as to ensure that contact with the locking face and impulse plane only occurs at the corner A of the tooth.

The exact inclination depends on the greater or less draw, and on the height of the inclined planes; for this regulates the extent of the angular motion of the lever, a movement that determines the pitch of each pallet, that is, the distance to which it travels within the circumference of the wheel.

It is very easy, when drawing an escapement, to accurately fix the extent of this inclination; but in the great majority of cases all requirements will be satisfied by sloping the front face of the teeth at an angle of between 25° and 28° to the radius, say an angle B A D of 27° (fig. 3, plate VII.).

The teeth should be cut away behind so that the pallets may enter at the back without touching.

735.—Clubbed teeth should always terminate in an inclined plane as shown in fig. 3. If, instead of this being the case, the extremities retained their original form and were portions of the circumference, the drop would be increased by the entire thickness of the tooth and the watch could never be timed.

Jurgensen inclined them at an angle of 5°* but he does not explain whether this inclination is to be measured in the manner previously employed for determining the height of the impulse planes (**704**) or the extremity is to make an angle of 5° with a tangent drawn to the circumference of the wheel.

In the first case, the incline on the tooth having the same height as that on the pallet, there would be no locking at all and the front corner of the tooth would fall on the impulse face. On the second interpretation, the height of the incline would be so slight that, during the *second portion* of the half-lift (**711**), the tooth would pass by the lower corner of the pallet with very little pressure and considerable rapidity. Some of the impelling force would thus be wasted and go to increase the drop.

736.—If we observe that with club teeth the amount of the locking which, with a pointed tooth, is as much as 5°, is diminished by the incline of the tooth, and remembering that experience has proved a mean locking of 2° to be essential, it is evident that the height of the incline on the tooth should be rather less than half the 5° included between M and S (fig. 4, plate VII.).

* He says, "The inclination of the teeth of the wheel should be 5°." This phrase is objectionable in that it is not perfectly clear; but it is not likely that he was referring to the slope of the front face of the teeth.

The article on designing a lever escapement will explain the manner of tracing this incline, which should be straight.

Efforts have been made to determine what degree of convexity would be best suited to this incline, but such a research is utterly useless; for it is obvious, when we consider an escapement of ordinary dimensions, that any curve would only cause the incline to approximate more or less towards a portion of the circumference of the wheel, and thus to increase the energy of drop at the expense of the impelling force.

The Breadth of Pallets and Teeth.

737.—The breadth of the pallets should be half the interval between the extremities of two successive teeth when these are pointed.

When clubbed, the breadth of the pallet should be such that, if added to that of the tooth, the two together are exactly equal to half the interval between two teeth, in other words, between F and A or N and C (fig 3, plate VII.).

Thus, if the width of the pallet be identical with that of the tooth, each will be one-quarter the distance between the points of two teeth, $f' g'$ (fig. 1, plate VIII.).

These several dimensions should be made slightly less in order to give the requisite freedom and so avoid friction between the heels of the pallets and teeth.

The acting faces of teeth and pallets.

738.—If we examine a number of lever escapements both from above and edgewise, drawing a vertical section of the tooth and pallet engaged with it, we shall obtain the figures E, H and N of fig. 48.

The second, E, is that most frequently met with. It is objectionable because, if the middle of the tooth does not exactly coincide with the centre of the pallet, that is with the line *e*, the wheel will give an oblique pressure on the lever; and this has a very detrimental effect.

The operation of pitching and of setting the jewels is not always done with absolute accuracy, and some makers give the acting face of the tooth the flat form shown at H in order to avoid this source of irregularity. It has the effect of materially diminishing the fault when it exists.

The fourth form, shown at N, which is often met with in the best watches, is produced by cutting away the two corners of the face of the tooth, thus forming a beaded edge between two bevelled sides. (These bevels should be left gray and not polished, **81**.)

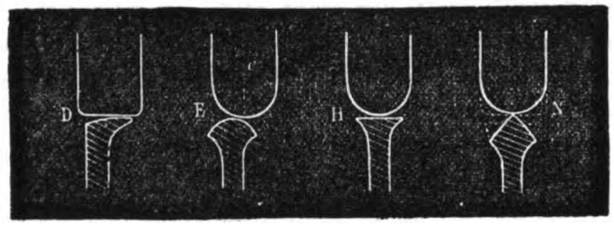

Fig. 48.

It is a recommendation of this plan that it retains the oil so that it flows back and is not driven from the points of contact, and the escapement is rendered somewhat less sensitive to the thickening of the oil; but it is essential, when the face of the pallet is rounded crosswise, that both escape-wheel and pallets be planted with care so as to avoid oblique pressure.

It is better to retain the form of the pallet rectangular, as shown in dotted lines at N or as at D.

Finally the two faces may be planes inclined to each other so that contact only occurs against a rounded edge of the tooth as at D. Such an arrangement has, in the lever escapement, advantages analogous to those of a Breguet cylinder escapement (**588**).

739.—With a view to retain the oil Breguet made a cut in the ends of the teeth; others have drilled a hole; others again have formed a double groove or sloping notch (shown dotted at N) inclined from the side that rests against the locking face. These devices effect the desired object but they are little employed since the results obtained are not commensurate with the difficulties of construction; these results are very little superior to those secured by an ordinary wheel that is well made, by a watchmaker possessing the requisite knowledge of the laws of hydrostatics (**81**).

Pointed Teeth and Teeth that terminate with an inclined plane.

740.—In France and Switzerland the teeth of lever escapements are *clubbed*, that is to say enlarged at their extremi-

ties and considerably contracted towards the base. In England pointed or *ratchet* teeth are preferred.

In the latter case the entire impulse plane is on the pallet.

With clubbed teeth about one half of the impulse plane is on the tooth and the other half on the pallet.

Lastly a third form has been tried in which the entire impulse plane is on the tooth. The pallet then becomes pointed.

All these arrangements are satisfactory, and good results have been obtained with them in the hands of intelligent horologists.

Impulse plane on the Teeth.

741.—When the entire impulse plane forms part of the tooth of the wheel the pitch on the locking face can be reduced to a minimum as well as the clogging produced in course of time by the thickening of oil. Moreover the extent of the plane and the fact of its being isolated on a thin support render it an easy matter to so adjust the plane as to avoid loss of time and to retain the oil; this is not so essential when the wheel is of brass but still it is an advantage.

This arrangement requires care and accuracy of construction; and a watchmaker desirous of experimenting on it should ascertain, in accordance with the laws of our Theory of Escapements, what forms are best suited both for the impulse plane of the tooth and of the pallet; the latter must not have a cutting edge. The action is precisely analogous to that of the incline of a tooth against the edge of a cylinder.

A beautiful specimen of this form of lever escapement, perfectly made, was shown at the Exhibition of 1855 by a modest but very clever watchmaker, M. Sylvain Mairet of Locle.

Escapement with Ratchet Teeth.

742.—In a small work on watchmaking we read; "Pointed teeth are best suited to the lever escapement because they traverse a longer impulse plane and therefore exert a greater force on the balance."

This assertion is utterly gratuitous.

For consider an impulse plane *n o d* (fig. 49): the pointed tooth *b* will impel it backwards through the interval *d h* and in doing so will traverse the full extent, *n o d*, of the plane. Now assume half of the plane to be on the tooth and half on the pallet so that we have the two inclines *n i* and *i d*. The corner *i* of the tooth will traverse the incline *i d*, and the corner *d* of

the pallet will be driven backwards over the plane *n i*. On the whole the displacement of the pallet will remain the same (*h d*), and it will be just as if a pointed tooth successively passed over the two planes *n i* and *i d* joined end to end, so that in fact their total length would be slightly in excess of that of the original plane *n o d*.

Fig. 49.

743.—An escapement with pointed or ratchet teeth has the following objections and advantages:

Both the pitch with the locking face and the drop are very nearly doubled; there is therefore an appreciable increase in the resistance opposed to unlocking, especially when the oil is at all thick. Out of the 10° through which the pallet moves, a greater proportion is expended in the unlocking (**705**). Lastly the fine pointed tooth must be made of brass, it is liable to wear and distortion and is ill-adapted for retaining oil, which must be applied in very small quantity.

On the other hand its advantages consist in: (1) the pallets having double width so that a greater quantity of oil is retained on them; (2) the escapement will go for a considerable time after the oil has gone bad or thickened. Some watchmakers indeed do not put any oil on either the teeth or pallets when the wheel is made of a particular kind of brass, known in France as English brass, but the point of the tooth wears in time; lastly, (3) the escapement is more easy of construction. When this form is adopted the escapement can be made with sufficient accuracy by ordinary workmen; for if the planes are inclined to the requisite extent there will be no time lost in the lift.

Escapement with the Impulse plane partly on the pallet and partly on the tooth.

744.—As compared with the ratchet toothed wheel, the wheel with clubbed teeth possesses the following qualities:—It retains the oil better;—The friction occurs at two points of contact instead of one;—The impulse commences with a shorter lever and is, therefore, more efficient;—No wear or distortion or varia-

tion of the acting surfaces need be feared when the wheel is carefully made and of good material;—It is possible, within certain limits, to reduce the pitch with the locking faces if necessary, and thus, while diminishing the effect of viscosity on these surfaces, to increase the *real lift* that corresponds to a given apparent lift (**706**).—Lastly, the drop can be reduced to almost nothing.

It is undoubtedly true that, as a set off against these advantages, it may be objected that this escapement is of a highly scientific character so that its construction is a matter of some delicacy and requires the skilled hand of a first-rate workman.

SUMMARY OF THE TWO LAST ARTICLES.

745.—The English lever, as usually met with, can be made by workmen of average ability, for its acting surfaces are somewhat extensive, escape-wheel easily cut, motive force considerable, and balance heavy, and yet it will possess a good rate, that is abundantly sufficient for the requirements of the public.

The French escapement demands more care and delicacy of construction and a more thorough knowledge of the laws of mechanics and hydrostatics; but when well made it gives a better and more uniform rate, that will remain the same throughout a very long period, since there is no danger of wear or distortion of the acting surfaces.

In conclusion, then, the advantage is on the side of the French lever. It is not fair to make this excellent mechanical arrangement answerable for the ignorance of those manufacturers, unfortunately far too numerous, who bring into the market such numbers of lever escapements that are contemptible either on account of their arrangement or workmanship, and far inferior to the average class of English watch.

The Drop.

746.—The amount of drop depends on the pitching of the wheel and pallets.

It is hardly necessary to observe that in the lever escapement no more drop should exist than is requisite to ensure the free action of the several parts; for, besides having an influence on the timing, every watchmaker is aware that long drops occasion a shake of the lever, thus causing it to come away from the banking and to remain locked without any point of sup-

port; in which case very injurious friction will occur between the safety-pin and roller. This being so, how happens it that in a great number of Swiss watches the drops are excessive? It arises from the fact that workmen do not as a rule take sufficient care to make the interval between the corners *b* and *c* (fig. 2, plate VII.) less than the distance between *b* and the resting point on the face *c n* of the pallet G, by at least the entire extent of recoil, etc.

SUMMARY OF THE PRINCIPAL PRACTICAL DATA.

747.—*The centre of the pallet-staff* must, for equal impulses, be at the point of intersection of the two tangents to the wheel drawn at the extremities of the radii that pass through the middle points of the impulse planes of the pallets.

When the impulses are required to be in proportion to the resistances, it will be at the intersection of the tangents drawn through the entrance edges of the locking faces.

Opening of the pallets (measuring the interval between the two locking faces or between the middle points of the impulse faces):—two spaces and a-half, or 60° for a wheel of 15 teeth and 75° for one of 12 teeth, etc.

Draw:—15° on the engaging and 12° on the disengaging pallet.

Height of the impulse planes of the pallets:—from 5° to 6°, that is to say, such as will produce an angular movement of the lever of 10° to 12° at most (the smaller amount is preferable).

These inclines should be straight.

Height of the impulse planes of the teeth:—This is determined by the space that corresponds to about 2° or 3° taken from the 5° of locking.

Total Lift:—between 45° and 60°, varying inversely with the size of watch and directly with the resistance, inertia and friction.

Inclination of the front face of the teeth:—26° or 27° as a rule. The teeth must be cut away at the back.

Thickness of the pallet and tooth taken together is exactly half the interva between the point of one tooth and the corresponding point of the next succeeding tooth (allowance being made for freedom).

In factories the average *distance between the centres of the pallet and balance-staff* is made equal to the diameter of the wheel.

The length of the fork arm of the lever should be:

For a lift of 40°, 4/5^{ths} the distance separating the centres.
„ 45°, 9/11^{ths} „ „ „
„ 50°, 5/6^{ths} „ „ „
„ 60°, 6/7^{ths} „ „ „

Diameter of ruby-pin:—about a third the distance between the points of two teeth.

This pin should be either triangular or oval and flattened on its front face.

For details as to the size of roller and the position of the ruby-pin see articles **731** and **716**

TO DESIGN THE LEVER ESCAPEMENT AND TO CALCULATE ITS PROPORTIONS.

748.—If the reader has attentively followed the preceding considerations, he will understand that in this escapement as in others, and even in a more marked degree than is the case with these others, all the several dimensions are correlated. The slightest change in one of these proportions will disarrange the harmony of the whole and render necessary a readjustment which, though at times only partial, usually extends to every part. On this harmony depends the accuracy of the results obtained; but it must not be forgotten that when dealing with such microscopical quantities, whose reciprocal actions are often unknown to us, the experience and intelligence of the workman have to be relied on to supply various practical details that could not possibly be enumerated in an elementary treatise. At the same time, however, we can assure every intelligent reader that after *studying* this work, not merely reading it in a rapid and superficial manner, he will find that all those every-day difficulties, which have caused him so much trouble, cease to exist.

It necessarily follows from the condition of the entire escapement being in harmony that, when one of the principal elements is known, it is possible to deduce all the others, so that it only remains to construct the escapement in accordance with previously ascertained dimensions.

When results of exceptional accuracy are required, such calculations involve the use of trigonometrical formulæ that are unfortunately beyond the reach of a great number of readers; we shall, then, in nearly all cases employ the approximate practical method as we have hitherto done, and will subsequently give the mathematical solution. Those in a position to follow it will derive considerable advantage; but it is advisable, before discussing these methods, and in order to facilitate their use, to design the escapement on a large scale. Such a drawing if well made will give all the proportions of the escapement with sufficient accuracy.

To Draw the Escapement.

Plan of the Pallets.

749.—We shall pass rapidly over certain details, as those that have already been given in the discussion of the duplex (**548**) will be fresh in the reader's mind.

The pallets we proceed to draw form part of what is known as a *right-line escapement with visible pallets;* the equipoise of the pallets and lever, which is always necessary, is most easily secured with this form of escapement.

It is one of those in which the lifts are equal, that is to say, the middle points of the impulse planes are tangential. The lifting arc is 50°.

Let the escape-wheel have 15 teeth and be 9 mm. in diameter.

9 millimetres multiplied by 10 gives 90 mm. (3·5 ins.) for the diameter, and therefore 45 mm. (1·75 ins.) for the radius.

From the centre A and with a radius of 45 mm. draw the circumference *f g k i* of the wheel (fig. 1, plate VIII.).

Draw the two radii A D and A E, inclined at an angle D A E of 60°, the opening of the pallets (747) for a wheel of 15 teeth.

At the extremities of these radii draw the two tangents D C and E C; their point of intersection is the centre of the pallet-staff.

Draw the line of centres A C B through the two points A and C.

The width of the pallets is as we have already seen dependent on that of the teeth. Assume them to be equal, so that each will measure 6° (one-quarter of 24° or the distance between two points).

Mark the width of the pallets lightly on the circumference of the wheel, remembering that the lines A D, A E, must coincide with the middle points of these pallets. The points *a* and *e* are thus determined.

From the centre A of the wheel draw, through *a* and *e*, the two lines A G, A F; then through *a* draw the line *a* I making an angle G *a* I of 15° with *a* G; and through *e* draw *e* J inclined at an angle F *e* J of 12° with *e* F.

The two lines *a* I, *e* J will mark the locking faces of the pallets and, in order to determine the opposite sides of these pallets, it will be only necessary to draw the two lines *b* V, *d* L parallel to them through the points that fix the width.

Draw the lines C W, C H, cutting the line V *b* in *b* and L *d* in *d*, and forming with the tangents C D and C E two angles E C H, D C W, each equal to 6°*, and so fixing the height of the impulse planes of the pallets.

* In order to avoid crowding and confusing the diagram these angles are made to measure 6° instead of 5°. The additional degree makes up for the loss of lifting action referred to in (704) and indeed for that due to the shake of the pinions; as a matter of fact, the angular movement of the lever will never be more than 5°.

After joining *a* and *b* and *e* and *d*, the pallets may be inked in as well as such dotted lines as it may be desired to retain.

To draw the wheel.

750.—From the point *a* draw *a* Q making an angle Q *a* A of 25° or 28° with the radius A *a* of the wheel. The line Q *a* gives the inclination of the front face of the tooth. (If the exact amount of this inclination is required it will be necessary to draw the disengaging pallet in the position of locking, the tooth resting against the locking face. The front of the tooth must then be inclined to the extent requisite to prevent adhesion.)

The interval between each two teeth being 24°, mark round the circumference of the wheel, commencing from the radius A *a*, the points $f', g', k', i', j', l' p', q'$, separated by 24° from each other; then, behind these points at a distance of 6° (or the thickness of a pallet) mark the other points f, g, k, i, j, l, p, q. The points f', g', etc., give the positions of the front faces or points of the teeth and f, g, k, etc., indicate the heels.

To draw the front faces, from the centre A draw the circle to which Q *a* is a tangent; it will then only be necessary to draw tangents to this circle through the points f', g', i', etc., and these will give the inclination of the front faces.

Since the angle H C P indicates the angular movement of the lever, that is to say the depth to which the point of the pallet enters within the circumference of the wheel, the line C P fixes the depth of the space between two teeth; this amount is always exceeded when scribing out the rim of the wheel in order better to ensure freedom.

It will be seen from the engraving that the radius of this rim is one-fifth or one-sixth less than the total radius of the wheel.

From the centre A of the wheel describe the dotted circumference $f' g' k'$, etc., which should overlap about half the impulse planes on the pallets (more or less according as it is intended to increase or diminish the pitch with the locking faces). The distance between the two dotted circles indicates the height of the small inclines formed on the teeth. They may be drawn in, as well as the back faces of the teeth, which should be much cut away, in order to avoid contact with the pallet, *d*, when in motion.

All the portions of the drawing that it is desired to retain may now be filled in with ink.

To draw the lever, roller, etc.

751.—From the point C, the centre of the pallet-staff, and

with a radius equal to the diameter of the wheel, mark off the point B, the centre of the balance. (This is a mean between the dimensions in general use. The watchmaker must decide for himself as to whether it can be modified with advantage, **727**.)

Draw the lines C M, C N, inclined at angles of 5° on either side of the line of centres C B; the total angle M C N will thus be 10°.

From the centre B draw the two lines B R, B S, each forming an angle of 25° with the line of centres, and therefore enclosing an angle, R B S, of 50° (**701**).

With B as a centre and the radius B *o* describe the arc T T'. The point at which this cuts the line of centres gives the middle of the ruby-pin and this may now be drawn in, making its greatest diameter rather more than the width of a pallet (**723**), and its least diameter two-thirds the greatest.

From the centre of the pallet-staff, C, with a radius C *o*, describe the arc *o z x*, which determines the length of the lever measuring to the corners of the horns.

The notch in the lever is of such a depth as to guarantee perfect freedom in the action of the ruby-pin.

The circular arc R S, passing through the base of the notch, fixes the position of the point of the dart (indicated by dotted lines) or of the guard-pin, and the point of intersection of R S and C M, that is *u*, gives the size of the roller.

In the case we are considering, however, the guard-pin is replaced by a projecting tongue, and instead of a table-roller there is a safety-roller.

With the centre B and a radius equal to two-thirds the length B *x* describe a circle to represent the size of the roller (**732**). This circle cuts the line C M in M. The arc M *r* N passing through this point and described with the centre C determines the position of the point of the projecting index-piece.

When the balance-staff carries a table-roller, the size of the passing-hollow is ascertained by drawing from the centre D (fig. 3, plate VIII.) two radii, *m* and *n*, passing on either side of the ruby-pin through the points of it by which contact is last made.

Its depth will be determined by allowing a sufficient interval of safety beyond the point of the dart.

As regards the width of the passing-hollow it must not exceed the lifting angle (*o* B *x*, fig. 1).

In order to ascertain the angle between the two arms X, X', in escapements in which the lever, during the locking, banks

against the axis of the escape-wheel, two lines, C Z, C O, are drawn from the centre C (fig. 1) of the pallet-staff, tangential to the axis of the wheel (which is at times made somewhat larger at this point), the pallets occupying the position indicated in the figure; then draw two other lines C *v*, C *y*, making the angles *v* C Z, *y* C O, with the first pair of lines, each being equal to an angular movement of the lever during a half-lift; say 5°.

The angle *y* C *v* gives the interval between the arms.

Two observations remain to be made.

When the ruby-pin is carried on a small projecting arm, this should always be prolonged so as to form a counterpoise and prevent any disturbance in the equipoise of the balance; the form indicated at *z z'* satisfactorily answers the purpose.

The internal faces of the horns should not be struck from B as a centre, but from two points to the right and left of B and indicated in fig. 1 by the small figures 1 and 2. The point 1 serves as a centre for the inner face of *o*, and 2 for that of *x*.

Their radius of curvature should be such that, when the escapement is in the position indicated in fig. 3, the concave surface of the horn is approximately parallel to the path of the ruby-pin.

In practice the radius is usually equal to the line B *o*.

Having now decided upon all the important dimensions, it only remains to draw in the lever, giving it as graceful a form as is possible, but at the same time taking every precaution to ensure that the piece may be evenly poised.

The escapement will thus be complete, and well proportioned in all its parts.

If now we assume the wheel to be set in motion, the tooth *k k'* (fig. 1) will move up to the impulse plane *a b*, force it backwards through an interval of about 5°, and, at the termination of this half-lift, the tooth *j j'* will fall against the locking face of the pallet E, which by that time will have moved inwards towards the rim of the wheel as far as the line C P. The acting surface of this locking piece will be included within the angle H C E, and, since the small incline on the tooth reduces the amount of the locking by 2°·5 or 3°, the actual pitching of the tooth and pallet will only be 2° or very little more.

The passage of the tooth in contact with the pallet D causes a movement of the lever of 5°, so that the middle of the lever will be brought on to the line C N; in moving to this position it will have driven forward the ruby-pin so that it emerges from the notch,

leaving the fork in such a position that it can re-enter this notch on its return, without coming in contact with the horn (fig. 3).

The lever will be compelled to remain against the banking by the roller passing with very little freedom in front of the dart or guard-pin, etc.

To calculate the Proportions of an Escapement assuming only the diameter of the Escape-wheel to be known.

Employing only the Elementary Rules of Arithmetic.

752.—Assume the wheel to have 15 teeth and to be 8 millimetres (0·315 ins.) in diameter and take 0·1 mm. (0·0039 in.) as a unit of measurement.

The diameter of the wheel will then be 8 multiplied by 10 or 80 *tenths*.

Multiply this latter figure by 3·14*; this gives 251·20 as the circumference of the wheel.

The circumference (251·2) divided by the number of teeth (15) gives as a quotient 16·74; this then is the interval between the points of two teeth.

The two points are then 16 tenths and 74 thousandths of a millimetre apart or, in round numbers, 1·67 mm.

Since the space between two points is equal to twice the width of a pallet plus twice that of the head of a tooth, and since the tooth and pallet should be of equal thickness, it follows that 1·67 mm. divided by 4, or 41 *hundredths* will be the thickness of each pallet and tooth and, moreover, of the cutter employed for cutting the pallet arms in order to insert the ruby pallets.

The interval between one point and the next comprises a tooth and space; the thickness of the cutter for making the teeth of the wheel should then be three-fourths of 1·67.

It is, however, impossible to cut a wheel satisfactorily by means of a single cutter. It must be done in several rotations, or at once by employing a double cutter (fig. 11, plate VIII.) whose two acting faces are separated by an interval equal to that between the face of one tooth and the heel of the next but one in front, that is by 2·93 mm.

The spaces between the teeth (**750**) should have a depth equal to about 1/5th the radius of the wheel, and thus the diameter of the rim will be 80 tenths minus 1/5th of 80, or 64 tenths at most, and the depth of each space will be 8 tenths.

* Since the geometrical ratio of the diameter to the circumference is, as we have seen (**137**), 3·141592, it follows that if the diameter is 80 the circumference must be 80 × 3·14. . . .

The opening of the pallets (**687**) includes the points of three teeth and half an entire interval or, in other words, two and a half intervals, and is therefore equal to 16·74 tenths multiplied by 2·5, that is 4·18 mm. along the circumference.

The width of the pallet should equal that of the tooth, that is it should be a quarter the distance between the points of two teeth ($\frac{16·74}{4}$) or, approximately, 4 tenths; the extreme diameter of the ruby-pin may then be 5 tenths and the least diameter rather more than 3 tenths.

753.—If, instead of assuming the circumference to be divided in tenths of a millimetre, we assume it to be divided into degrees, it will be evident that, for a 15-tooth wheel, the opening of the pallets should be 60°, the thickness of each pallet 6° and that of a tooth 6°.

Thus for a 15-tooth wheel 80 tenths of a millimetre in diameter, and, therefore, having a radius of 40 tenths:

The opening of the pallets is	4·18mm.	or 60°.
The breadth of each pallet	0·41	„ 6°.
The breadth of each tooth	0·41	„ 6°.
The thickness of the cutter for slitting the pallet arms	0·42	
The thickness of the double cutter for forming the teeth	2·93	
The depth of the spaces about	0·80	
The extreme diameter of the ruby-pin	0·50	
The least diameter of the ruby-pin	0·30	
The size of the notch in the fork allowing $\frac{2}{10}$ths freedom	0·70	

In considering the practical details connected with the lever escapement we shall give the method to be adopted for determining the several dimensions that are not mentioned in the above table; such as the height of the impulse planes, the position of the ruby-pin and guard-pin, the length of lever, diameter of roller, etc., etc. These can be ascertained without difficulty by carefully drawn lines on the calliper of the escapement, which will render it easy to rough out the whole with sufficient accuracy.

To calculate the proportions by the aid of Trigonometrical Formulæ.

754.—The preceding method will suffice for the great majority of watchmakers; but for the benefit of those familiar with trigonometry we proceed to explain a mode of calculation that is both more exact and more complete than that just given.

We will assume, as before, that the wheel has 15 teeth and is 8 millimetres in diameter and take the tenth of a millimetre as a unit of measurement.

It is essential throughout the calculation always to retain a sufficient number of decimal places when very accurate results are required.

The early portion of the process is precisely similar to that already explained.

The diameter of the wheel is, then, 8 × 10 or 80 tenths.

The circumference is 80 × 3·1416 = 251·32. This circumference divided by the number of teeth ($\frac{251\cdot32}{15}$) gives the interval between the points of any two consecutive teeth.

This interval being known, ascertain from it, as before, the thickness of the pallets, of the tooth and of the cutters employed to slit the pallet-arms, and to cut the wheel.

The radius of the wheel is 40 and this number divided by 5 gives the depth of the spaces.

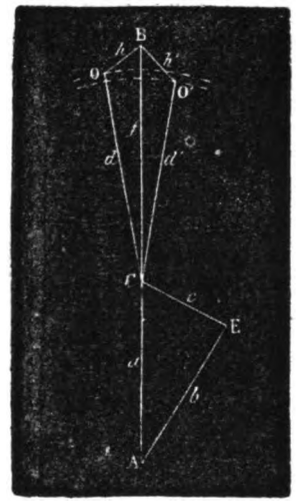

Fig. 50.

The triangle A E C (fig. 1, plate VIII. and fig. 50 in which the lettering is identical) is rectangular, having one side and the three angles known. The side A E (or the radius of the wheel) is 40. C E A is a right angle. C A E is half the angle of the opening (30°), so that A C E is 60°. We have then, in fig. 50, representing each angle by the letter at its summit, E = 90°, A = 30°, C = 60°, b = 40, and by Trigonometry

$$Sin\ C = \frac{b}{a}$$

$$\text{and } Tan\ C = \frac{b}{c}.$$

Resolving these equations we shall be enabled to find the value of a, or the distance between the centres of the wheel and pallet-staff, and of c, or the length measured from this latter centre to the middle of the pallet, in other words the point of intersection of C E, A E (fig. 1, plate VIII.), which may now be fixed by striking two intersecting arcs with the compasses.

The distance between the centres of the balance and pallet-staff is not required to be any definite quantity, but is usually made equal to the diameter of the wheel (**751**). Let us adopt this figure.

Examining the triangle C o B (fig. 1, plate VIII.) we see that the side C B is known (80), and that the angle B C o is the angular movement of the lever, say 5°; the angle o B C is half the total lift (say 25°), and therefore the angle B o C measures 150°.

In fig. 50, where this angle B o C is represented by B O C, we then have: C = 5°, B = 25°, O = 150°, f = 80.

And the proportions:

$$Sin\ O : Sin\ B = f : d$$
$$Sin\ O : Sin\ C = f : h.$$

will give the value of d (o C in fig. 1, plate VIII.) or the length of the lever measured from the centre of pallet-staff to the commencement of the horns, and the value of h (o B in fig. 1) or the distance between the centres of the ruby-pin and balance-staff.

Having determined the dimensions of this ruby-pin as already explained, the depth of the notch in the fork can be estimated, and, deducting this amount from d (fig. 50), we can form a new triangle B O' C of which one angle and two sides are known, namely C = 5°, f = 80 and d' = d minus the depth of the notch. (We here assume that the guard-pin is on a level with the base of the notch.)

Resolving this second triangle we obtain the value of h', that is the radius of the roller.

The several steps in the process are

$$Sin\ O' : Sin\ B = f : d'.$$

$$Sin\ O' + Sin\ B : Sin\ O' - Sin\ B = f + d' : f - d'.$$

$$Sin\ O' + Sin\ B : Sin\ O' - Sin\ B = tan\ \tfrac{1}{2}(O' + B) : tan\ \tfrac{1}{2}(O' - B).$$

and, when o' has been determined,

$$Sin\ O' : Sin\ C = f : h'.$$

All the proportions of the escapement have thus been accurately determined with the exception of the heights of the inclines, which can be so easily fixed upon practically that it is quite useless to resort to a complicated trigonometrical problem, that would involve considerable difficulty in its application, owing to the smallness of the angles and surfaces under consideration. The same may be said of the passing-hollow in the roller.

The dimensions obtained by calculation can be easily transferred by employing either a micrometer or a proportional compass.

Having once determined the relative dimensions of an escapement, it is possible, by assuming the diameter of the wheel to be either 1, 10, or 100, to ascertain the relative sizes of the several parts of any other escapement providing that we know the diameter of the wheel and the number of its teeth.

CHAPTER III.

CONSTRUCTION AND REPAIR OF THE LEVER ESCAPEMENT.

Practical details relating to the Escape-wheel.

755.—The wheel may be made of either steel or brass.

A brass wheel is, however, better than one of steel; and we can here only repeat what has been already said in article **550**. We refer the reader to it, and would point out that it is by no means a very rare occurrence to meet with a lever escapement with a steel escape-wheel in which the locking faces or impulse planes are pitted, in consequence of the mechanism having gone for a long time after the oil on the pallets had dried up. This results mainly from the employment of Balas or Spinel instead of Oriental rubies, the only kind that ought to be used.

When making an escape-wheel, the brass known in France as *laiton à la croix* must not be adopted until it has been satisfactorily experimented on, although it is recommended in a manner far too positive by many makers; neither is it well to adopt any of those kinds of brass that are very malleable, for they require a prolonged hammering and should be much reduced in thickness by the hammer in order to acquire the requisite hardness. Brass of this quality bends when the teeth are cut and thus the wheel is distorted.

The workman should select a quality of brass that is some-

what brittle, in other words one that is rapidly hardened under the hammer and does not require to be very materially reduced in thickness.

These details will suffice for every intelligent watchmaker; his experience and discretion can alone guide him in the absence of very exhaustive directions, such as cannot find a place here.

In factories the escape-wheel is made of a size that is quite empirical, being primarily determined by the radius of the fourth wheel. It is important to remember what has already been said as to the gradual decrease in the radii or levers of the train (**345**, **399**), a decrease that should be somewhat more rapid in the case of a small watch, for reasons analogous to those given when considering the cylinder escapement (**400**); but, assuming all the proportions to be alike in two watches that have the same general dimensions, one being a cylinder and the other a lever, the diameter of the lever escape-wheel should be slightly less than that in the cylinder. Experience and a study of well-proportioned watches will indicate to the maker how much this difference should be. It might, approximately, be ascertained by calculation but the discussion of this method would lead us beyond the limits of an elementary treatise.

In English watches, where the balance is larger and the vibrations slower than in those of French make, the diameter of the escape-wheel is made about one-sixth less.

To make the Escape-wheel.

756.—The lever escape-wheel presents fewer difficulties of construction than a duplex.

A steel wheel with clubbed teeth may be made with ordinary cutters, but a pair of these are necessary; one, cutting with its edge, to form the straight faces of the teeth and to cut out the spaces, and the other cutting with its edge and side in order that it may round the back of the teeth. These two cutters may be rigidly connected together (fig. 11, plate VIII.).

Every watchmaker will be able after a few trials to prepare his own cutters.

The steel should be first thoroughly annealed and care should be taken in cutting the wheel only to remove a small quantity of matter at a time, to keep the cutter supplied with oil, and to clean it from time to time with a scratch-brush.

A brass wheel may be cut either with a single rotating cutter, a hooked cutter or by one of the ordinary form.

As in the case of a duplex wheel, the entire mass of metal between two teeth must not be removed at once but by several successive operations.

To ascertain the position in which to set the cutter so that the front face of the tooth may be made of the required inclination, a disc of the same diameter as the wheel must be fixed on the table of the cutting engine. Having set the point of the index in a division in the 360 circle and fixed a sharp pointed cutter on the arbor, so setting it that its point corresponds accurately with the axis of the table, that is, with the pump centre, a mark is made on the edge of the disc; after moving the index over a number of divisions corresponding to the degrees of inclination of the face, a second mark is made on the edge. The interval between these (o A, fig. 10, plate VIII.) gives the distance at which the face of the cutter that forms the straight side of the tooth should be set on one side, in other words it is the distance of this face from the perpendicular drawn to the axis of the cutter and passing through the centre of the table.*

The small inclines at the extremities of the teeth are usually formed on the cutting engine, employing a cutter whose edge is inclined to the same extent as the planes are required to be. They can also be formed on the tool used for making the inclines of cylinder escape-wheels or some analogous appliance. Some such method is best for those who but rarely have occasion to make an escapement, since it is then possible to test the inclination and to adjust it so as to give the required depth on the locking face. It moreover has the advantage of enabling us at the same time to true the circumference of the wheel.

This incline of the teeth is made straight in the direction of its length but is very frequently rounded crosswise. (See **738** and the following articles.)

When the inclination that should be given to these inclines is not known with sufficient accuracy to enable the workman to finish them, he must wait till after completing the pallets. A verification on the depthing-tool will enable him to determine the exact degree of inclination that is needed.

It is a mistake on the part of the Swiss workmen to polish the bevel of the teeth; fine smoothing is preferable, for the oil will adhere more firmly and will be retained in greater quantity

* The angles o D A (of 27°) and D o B (fig. 10) are equal since they are internal and opposite angles.

on its surface and more easily distributed; it will therefore keep the acting surfaces lubricated during a longer period (**81**).

A brass wheel can be hardened by a suitable annealing. For details of execution that are common to the lever, duplex, cylinder and, chronometer escape-wheels, see the articles specially devoted to these latter.

To design and execute the Pallets.

757.—The lever and pallets are sometimes made of one and the same piece of steel; but generally they are separate and maintained in their required positions by a screw thread cut on the staff and by steady-pins. In this case we might make the lever of brass, and thus avoid the necessity of placing oil on the ruby-pin. When the lever is of steel, in the best class of escapements, it is a frequent practice to face the notch of the fork with gold; but, even when this precaution is taken, it is always wise, according to the experience of our best makers, to provide both it and the ruby-pin with oil.

In factories a distinction is drawn between *visible ruby pallets* such as are seen in fig. 2, plate VII., and fig. 1, plate VIII., and *covered pallets*. In these latter the rubies are out of sight, being let into notches cut in the pallet-arms. This form is less ornamental than the other and slightly increases the weight, but it has the advantage of more effectually retaining the oil on the acting faces, providing care is taken to allow the steel to project a little beyond the inclined planes, while, at the same time, seeing that the thickness of the rubies and pallet-arms is not in excess of that absolutely required. The oil is then retained in sufficient quantity in the grooves formed between the steel faces and the ruby.

The construction of covered pallets, retaining the lever light and conforming to the principles we have laid down, presents greater difficulties than does that of visible pallets. The showiness of this latter, which should influence the seller rather than the maker, together with the above facts, makes it preferred in the more expensive class of watches. Its construction from a mechanical point of view leaves nothing to be desired, and it is confided to the most skilful workmen, whereas those possessing less ability are employed in making covered pallets; hence it follows that these latter are generally so heavy, so badly made and in every respect so inferior to the others that they have only been used in very common work.

At the present day, in Switzerland, escapements are made by piece-work, that is to say each workman is specially skilled in the construction of one particular piece and confines his attention to it without regarding the others, or even the proportions they should bear to each other; indeed he is nearly always incapable of understanding such proportions.

One is solely engaged in cutting out the pallet-arms with the punching machine; another shapes and finishes them to a certain invariable pattern; another makes the pallet-staff; another the roller, etc. By adopting this system they are actually enabled to produce a complete lever escapement for a few francs.

To draw the calliper of a right-angled Lever Escapement.

758.—When about to make a pair of pallets it is in the first place necessary to draw the calliper of the escapement with very great care.

The distances between the centres of movement should be known.

On a smooth brass plate, perfectly flat and not less than 2mm. (0·079 ins.) thick, mark and drill a hole for the centre of the wheel, previously scribing out its circumference carefully (fig. 5, plate VIII.).

Assuming the wheel to have 15 teeth the angular opening of the pallets will be 60°.

Wax the plate (which we will term calliper No. 1) on the table of the wheel cutting engine, centring the wheel by the hole (*a*) on the pump centre. After setting the index in a division of the 360 circle, draw the line *a b* (fig. 5) employing a pointed tool in place of the cutter. Now rotate the dividing plate 30 divisions to the right and draw the line *a d;* and then turn the plate again 60 divisions to the left and make the mark *a c*. The angle *d a c* of 60° is the angular opening of the pallet (**759**).

Having removed calliper 1 from the table, mark the points of intersection of the lines *a d*, *a c*, with the circumference with very great care, and at these points *d* and *c* drill two fine holes.

Now replace it on the table using *c* as the centre. The index must be placed in a division of any circle divisible by 4 of the dividing plate and the line *a c* set perpendicular to the line of the arbor centres (which can be verified by causing the point of the tool to traverse the entire length of *a c*); then turn the table through a quarter of a complete revolution and draw the line *c b*.

Proceed in the same manner to draw the line *d b*, using *d* as

a centre. The intersection of these two lines *c b*, *d b*, gives the point *b* where a hole must be drilled, taking very great care to avoid displacing the centre.

If the position of the balance-staff is not given it must be determined, and a hole, *g* for example, drilled to indicate its centre. The calliper 1 is again fixed on the table (using *b* as a centre and setting the index in the 360 circle) and the line *b g* drawn, passing accurately through the point *g*. Count five divisions to the right and draw the line *b i*, and, after moving the table ten divisions to the left, draw the line *b j*.

The hole *g* must next be used as a centre and proceed in the same manner to obtain the lifting angle; that is to say, with the index in the 360 circle and the line *b g* in a direction perpendicular to the arbor of the cutter, move the dividing plate through as many divisions as there are degrees in the half-lift, say 25°; draw the line *g h* and, after moving it back through 50 divisions, draw the line *g l*.

From the centre *g*, using a small and well-made screw bar compass, draw the circle *k* passing through the point of intersection of the lines *b i*, *g h*. The point, *z*, where this circumference cuts the line of centres gives the position of the ruby-pin, and a hole must be drilled here.

With the centre *b* draw the arc *m n* which determines the length of the lever; and from the centre *g* draw the circular arc *v* to determine the extremities of the horns.

The concave faces of these horns must not coincide with the arc *v*, for they are drawn from two eccentric points situated above and below *g* (**751**).

These two points are usually determined by the intersection of an arc, described with the radius *b g*, with two small arcs, struck with the radius *g v* and the upper corners of the notch in the fork as centres. The lower point thus determined is used to strike out the upper horn and *vice versa*.

The calliper of the escapement is now very nearly complete, for we know: the planting points *a*, *b*, *g*; the position of the ruby-pin; the length of the lever, etc.; and, as a matter of fact, this is all we require. It is, however, well, because it both facilitates the succeeding operations and avoids confusing the lines on calliper 1, to prepare a second plate, which we will term calliper 2 (fig. 6, plate VIII.) and, after fixing it face to face with calliper 1, to drill the holes *a*, *b*, *g*, *d*, *c*, *z*.

The holes, *c*, *d*, are now enlarged until their diameter is equal to the thickness of a pallet arm; no difficulty will be experienced in this operation since we know the size of the escape-wheel (**752**).

Enlarge the hole for the ruby-pin (widening it, if necessary, so as to bring the two points of contact of this pin on the circular arc *k* in calliper 1), and make the size of the hole equal to the diameter of the ruby-pin. Taking the edge of this hole as a guide we may determine the position of the guard-pin or dart, and then the size of the roller (**731**), unless it is preferred to wait until the pallets are cut. It will be well when making the roller to keep it slightly larger than that determined upon, even though it be necessary to reduce it after verifying the escapement.

Taking *b* as a centre, draw the line of centres *a b* by means of the pointed cutter on the wheel cutting engine; then rotate the dividing plate so as to draw the line *b g* making a known angle with the first, etc.

In the majority of callipers, other than those for *straight line levers*, the perpendicular to *a b* at the point *b* passes through the centre of the balance as shown in fig. 6.

Having placed calliper 2 on the table of the cutting engine, taking *d* as a centre and the index in the 360 circle, draw the line *a d* R (a radius of the wheel prolonged); now move the plate through 15 degrees to the left and draw the line *d v*; and, finally, two lines parallel to this and touching the two opposite edges of the hole *d*.

Proceed in the same manner with the hole *c*, but the dividing plate must only be moved through 12 divisions.

Drill two holes P, Q. These holes, whose use will be presently explained, should be made just beyond the edge of the pallet-arms when completed, that is to say, they should fall in that portion of the metal that would be ultimately cut away.

With the help of these callipers, 1 and 2, the pallets can be made without much difficulty and, as will be seen, only a little care and dexterity are required.

Circles on the Dividing Plate that can be made available in place of the 360 circle.

759.—We have indicated in the preceding article that all the angular divisions should be made with the 360 circle; but since, in the majority of wheel cutting engines, such a circle

does not exist, we would point out that the following numbers (or multiples of them) may be used in place of it.

In fixing upon the opening of the pallets, the index may be placed in the 12 circle; the distance between two divisions is then 30°.

Two lines can be drawn at right angles to each other by using this same circle, since 3 divisions include 3 × 30° or 90°.

For the angle of 10° traversed by the lever, use the number 72; an interval then includes 5°.

Mark out a total lift of 40° on the 18 circle, each interval of which corresponds to 20°, or on the 36 circle with intervals of 10°, etc.

A lift of 45° requires a circle of 144 divisions, where each interval represents 2·5°.

For a lift of 50° employ the 72 circle, where each division is equal to 5°.

Lastly, the draw or inclination of the engaging pallet is given by the 24 circle, each division of which is 15°; or on the 48 circle with intervals of 7·5°: for the draw of the disengaging pallet use the 30 circle with divisions of 12° or the 60 circle where they are 6°, etc.

To rough out the lever and pallets.

760.—Having prepared a piece of steel of the proper thickness and large enough to cover the holes *d*, P, Q, *g*, *c*, (fig. 6, plate VIII.), draw the line *x y* on it (fig. 8) and, on this line, drill the centre hole *r* in a convenient position. The steel plate is now held against calliper 2 (fig. 6), an arbor or perfectly round pin passing through *r* in the steel and *b* in calliper 2, and, when the line *x y* has been brought to coincide accurately with the line *b g* (calliper 2, fig. 6), the two are firmly held together by clamps or any convenient means, and the holes *d*, *c*, *g*, *z*, Q, P, are first pointed through and finally drilled in the steel plate on the drilling tool.

These holes should be precisely of a size with those in calliper 2.

Prior to removing the future lever and pallets from the calliper, describe, with the centre *a* and the radius of the escape-wheel, an arc of a circle, and draw lines corresponding to the two pairs of parallel lines touching the holes *c*, *d*. The lines thus drawn on the metal will serve as guides and marks for checking the work during the subsequent process of cutting the pallets.

Cut away the front of the steel plate by means of a file to the form shown in fig. 8, taking particular care however to remove rather less than half the holes, in other words only the amount included between the parallel lines indicating the pallet arms.

The pallets are now ready to be cut.

To cut the pallet-arms, prepare a pivoted disc, etc.

761.—Pallets are usually cut on a wheel cutting engine specially arranged for the work; but as the majority of workmen have not access to such an instrument we will proceed to explain another method which is very simple and can be practised without difficulty. Those who thoroughly grasp it will, after a trial of this method, be able to make for themselves, if needful, the additional appliances that are required for cutting pallets in a cutting engine.

In the absence of such a suitable tool, the pallets may be cut on the depthing tool; a small accessory piece which we have termed the *pivoted disc* is in this case necessary, and the following is the mode of making it for the case under consideration.

Take a steel plate about 2mm. thick and give it the form shown in fig. 7, plate VIII. Carefully turn the two points and make the cylindrical arms of equal diameters; then smooth the upper face of the disc so that it is level with the arms on either side.

Draw the line *c d* passing accurately through the points; and on this line at the centre of the disc drill a hole *s*.

Having waxed the disc on the table of the wheel cutting engine (using *s* as a centre) ascertain that the line *c d* is placed in a direction exactly perpendicular to the axis of the cutter arbor, in other words that, when the dividing plate is held at zero on the 360 circle and the point of the cutter coincides accurately with that of the pump centre, this cutter will throughout its entire path traverse the line *c d*.

After making this verification, move the index through 42 divisions of the plate towards the right and draw the line *b n*; bring the plate back to its zero position and then turn it through 15 degrees towards the left so as to draw the line *a m*.

These two lines should be continued across the edge of the disc in the manner indicated at *b* and *m*, fig. 7.

The disc is now removed and fixed on calliper 2 by means of a pin or arbor passing through the hole *s* of the disc and the centre *b* of the calliper. When the line *b n* of the disc coincides

accurately with *b g* on the calliper fix the two superposed pieces in position by clamps or, preferably, by a small hand vice faced with brass, and drill the hole *e* in the disc corresponding to Q in calliper 2.

Proceed in the same manner to obtain the hole *f* (fig. 7) which should correspond to P (fig. 6) when the line *a m* (fig. 7) corresponds to *b g* (fig. 6).

Having cut the two notches in the disc that are required for the passage of the cutter and smoothed the upper face, clearing it of all irregularities, slightly enlarge the three holes *s*, *e*, *f* (fig. 7), and fix in them three pins of blue steel turned true and projecting above the surface of the disc by rather more than the thickness of the pallets and lever.

The pivoted disc* is now complete; place the pallets on it (D, fig. 7) and fix them with shellac; then place the disc between the centres of a depthing-tool which must be adapted for being held in the vice.

The arbor carrying the cutter is placed between the other pair of centres and can easily be brought to the required position since the parallel lines on the pallets should fall accurately on either side of the cutter; and, moreover the half circumference of the hole that is not removed forms a very safe guide (**752**).

As the diameter of this hole is equal to or slightly greater than the width of a pallet-arm, the cutter should just pass easily into the space.

When the notch for the disengaging ruby pallet has been cut, remove the pallets from the disc and wax it on afresh; its centre hole is put on the pin *s* (fig. 7) and the hole *i* on the pin *f*, and then proceed as in forming the other side.

The line *x y* will serve as a guide to ensure that the two notches are cut of equal depths.

The pivoted disc can, if needful, be used for cutting pallets of very different sizes; and only the central pin need be retained, the other two being suppressed providing they are replaced by screws so situated that, when the pallets are laid on the disc with the line *x y* (fig. 8) coincident with *a m* or *b n* (fig. 7), the

* The triangle A B C (fig. 9) is equiangular. The angle B C A being equal to 60°, D C H will be 120°. Considering now the triangle D H C, we know that C D H is 12°, so the angle C H D must be 48°.

A perpendicular drawn from the centre M on the line H N will give the rectangular triangle M N H, in which M H N measures 48°; H M N is therefore 42°.

Similar reasoning would show that *a* M *b* must be 15°.

mere turning of these screws will suffice to firmly clamp the pallets upon the disc. (If any fear is entertained lest the arrangement be disturbed, a little shellac may be run round the pallets after they are clamped.)

The angles here prescribed are those ordinarily employed; if greater accuracy is desired in the inclination of the locking faces, they must be previously determined by means of a large scale drawing.

Finishing the Pallets and Lever.—To make the Roller.

762.—After cutting the pallets, describe, with the hole *y* (fig. 8, plate VIII.) as a centre, the small circular arc *v* (figs. 5 and 8), which determines the length of the lever and the starting points of the internal curved faces of the horns. The hole *w* (fig. 8) has a diameter equal to that of the ruby-pin, and it will therefore only be necessary to slightly extend this hole towards the centre *r* and square it up, to complete the notch of the lever. If a more accurate form is desired it may be finally shaped by means of a cutter.

There is no occasion to remove all the metal within the small circle *v* (fig. 8), but it should be cut away up to two short arcs struck within this circle, with its own radius and from excentric points above and below the hole *y*, as indicated at *g*, fig. 5. The lower point serves as a centre for the upper horn and *vice versa*. A slight excess of metal should be left, and the final adjustment made as needed, after having tried the escapement.

When the lever notch is completed, its depth will give the position of the point of the dart or the safety-pin, and it only remains to finish the lever, reducing it with a file with very great care. No more metal should be left than is necessary either for solidity or for maintaining it in equipoise when balanced on its staff.

By placing the pallets on calliper 1, with a pin passing through the two centres, and causing the lever to traverse its angle of 10°, the workman can ascertain that the metal on either side of the slots for the pallets does not approach so near to the wheel as to draw oil from its teeth.

When the lever is ready for hardening, the distance between its centre of movement and the extremity of the dart is measured by a compass, and, with the centre *b* in calliper 2 (fig. 6), a mark is made at this distance on the line *b u*.

The circle described with the centre *g* and passing through the point of intersection *u* fixes the size of the roller.

This roller (which should always possess a small central steadying collar formed by diminishing the thickness towards the circumference), after being roughed out to nearly the requisite size, is fixed by means of an arbor and clamps against calliper 2, and the hole for the ruby-pin is drilled.

A brass pin is now cemented in provisionally, and temporary brass pallets are also fixed in position; these should be slightly longer than necessary.

When thus far completed the pallets are placed on the calliper, an arbor passing through the two centre holes, where it is held by friction; the line *x y*, corresponding to the middle of the lever, is brought over the line *b g*, and, when the pallets have been fixed in this position, the circumference of the escape-wheel is traced with the fine point of a compass on the brass pallets, using *a* as a centre.

Turning the pallets on their centre, bring the line *x y* over *b i* of the calliper, and mark the circumference of the wheel on the disengaging pallet.

Now bring them backwards until *x y* coincides with *b j* and mark the circumference on the engaging pallet.

The interval between the two marks on each pallet gives the height of the inclined planes; they are formed by hand, taking care to leave a slight excess of metal, which will be subsequently removed as required after verifying the escapement.

To accomplish this verification, adjust an arbor firmly in the hole *a* of calliper 1. This arbor will serve as an axis for the wheel and, in the same manner, a second arbor, passing through the hole *b*, can act as an axis for the pallets; and the two mobiles, when thus held against the surface of the calliper, can be guided by the fingers, tweezers or pegwood, and caused to act just as they will ultimately do in the watch itself.

As an additional security it is well to make a second verification on the depthing tool. We shall subsequently indicate the mode of effecting this (**778**).

Knowing the angular movement of the lever during the passage of a tooth in contact with the inclined faces of the pallets, remove the wheel and place the roller in position on calliper 1; that is to say, centre it on an arbor traversing the hole *g*. Now cause the fork and roller to act together, in order to make sure that the pin only comes in contact with the sides of the lever notch, and that it passes the horns with sufficient freedom.

In testing this point it must be remembered that, at the commencement of the lift, it is the ruby-pin that leads, whereas the lever impels the roller, etc., during the latter portion.

The pin should be made somewhat long in order that the eye can observe its action with ease.

When the roller is rotated on the arbor that serves as an axis, the lever being held at the extremity of its path (of 10°), this roller should just touch the guard-pin. It must be turned down to a convenient size and the passing hollow roughed out; a further verification must then be made on the depthing tool.

We have just observed that when the lever has arrived at the extremity of its path the roller should merely touch the guard-pin; but as a matter of fact the locking point of the lever, that is the banking, is set just beyond an angular movement of 10°, so as to allow a slight freedom between the extreme points of the inclines and the circumference of the wheel. The distance that the guard-pin will thus travel beyond the point limiting the 5°. of the half-lift will nearly always give sufficient freedom between it and the roller.

The explanations above given render it easy to determine the position of the bankings or the width of the hollow containing the lever (when such a form is adopted).

After verifying on the depthing tool, the roller is hardened and finished with some care; for the passing hollow may cause it to lose its shape, especially during the operations of smoothing and polishing.

The edge of the roller should be rounded and very highly polished, as, by this means, the friction that occurs when the guard-pin comes in contact with it will be materially reduced.

In hardening the lever and pallets, the precautions that are essential with every delicate piece, a cylinder escape-wheel for example, must be observed; for it is very important that the metal should not be distorted during the operation, since in that case the opening of the pallets would be modified. This would involve the use of ruby pallets of a length slightly differing from that indicated by the calliper, and, the escapement not being in strict accord with the drawing, the position of the planting points would be altered (**698**).

The pallets are tempered to a blue, and the edges of the lever notch and sides of the dart, etc., are polished with care: after the workman has fixed the brass pallets in position, he again

verifies the escapement in the depthing tool, in order to make sure that no part has been distorted in the hardening.

Precautions to be observed, etc.

763.—When making either the calliper or the pivoted disc or the pallets themselves, it is always essential to make the several holes with the drilling tool after they have been first *pointed* with as much care as is taken in pitching a wheel, and this is especially necessary when the hole is to be made at the intersection of two lines.

When one piece is superposed above another in order to drill corresponding holes, the drill used for pointing should be cylindrical except at the actual blade, and it should enter the hole that is already drilled freely but without play. The hole should then be continued to some depth with a fine drill.

The arbors that traverse the holes in the calliper must be perfectly round and firmly held by friction. Providing the thickness of the calliper is sufficient, there will never be any danger of deranging the centres, however many times these arbors are removed and replaced.

It is almost useless to add that the wheel, pallets and roller should go easily, but without play, on to the arbors passing through the calliper that serve as temporary axes.

When tracing the several lines, the workman should only drill the holes that are required as centres in the wheel-cutting engine; all other holes, indicating the direction of lines, should be merely pointed. They will be drilled after the lines that pass through their centres have been drawn, because a point is always a more certain guide than a hole, except of course when this latter is very fine.

We have indicated no method for drawing the lines and angles except that in which the wheel-cutting engine is employed: it is the only one that secures a high degree of accuracy. In the absence of any better means it would be possible to use steel templates previously prepared, with their edges inclined at the required angles; but such a practice is uncertain and the employment of the grammaire or divided plate is to be preferred.

This appliance is described in article **507** and illustrated at fig. 1, plate V.; it might be used for making the templates here referred to.

In conclusion, the making of a pair of pallets does not present any serious difficulty, but it involves absolute accuracy

in the several measurements and very *honest* workmanship. The workman who does not thoroughly understand the escapement, or who is not practised in delicate processes involving such strict accuracy, will experience some difficulty, even after making a good many trials; and even the clever and intelligent watchmaker, who has never made a lever escapement, must not count on being successful at first; success, for which he will not have long to wait, must nevertheless in all cases be the outcome of a certain number of trials.

OBSERVATION.

764.—The pallets we have been considering are said to have equal impulses.

If it be desired to set the locking points tangential, the two lines *a d* and *a c*, fig. 6, plate VIII., instead of passing through the centres of the holes *d* and *c*, must only touch their sides, so that one hole is placed within *a d* and the other beyond *a c*.

This one observation should be enough to enable any reader to draw the calliper of either class of escapement.

To make a straight-line lever.

765.—The calliper will be traced out precisely in the manner already explained; for it can only differ in the position of the centre *g* (fig. 5, plate VIII.); and this change will be just as though the entire system *h i j l* were pivoted on the point *b*, and the lines *b g* and *a b* made continuous.

Additional details would be quite superfluous after those above given and the general explanations in the article on designing the lever escapement.

To make a lever with covered pallets.

766.—A pair of covered pallets is made in the same manner, and the same tools are employed. The breadth and inclination of the pallet-arms are marked out on the steel, and they are then cut on the pivoted disc; but for this operation two cutters are required. These are firmly fixed on an arbor, with a collet between them whose diameter is less than that of the cutters, and its thickness is equal to that of the pallet-arms. The arms will thus pass in between the cutters, the metal being removed on either side (fig. 12, plate VIII.).

Or a single cutter, forming only one side of the pallet at a time, might be employed.

The pair of pallets is now firmly fixed with shellac on its

back edge, and the arms are slit by means of a cutter set in the plane of the pallets, to adapt them for receiving the rubies.

In the absence of a special tool any watchmaker will be able to arrange one for this last operation.

Lever and index-piece in one.

767.—Double roller escapements are provided with an index-piece fixed to the lever by a screw and steady-pin, as shown in fig. 1, plate VIII.

Such an arrangement is delicate, and therefore increases the price of the escapement; this fact has led to attempts being made to make the lever and index-piece in one.

We will briefly indicate the successive operations that this condition involves:

After roughing out the lever, taking care to leave its fork end sufficiently massive, the inner faces of the horns are formed by means of a double cutter as shown at *b* (fig. 7, plate VII.), which leaves untouched a tongue that projects between the two horns.

The lever is next turned on one side and held against a cutter, *c*, which removes the metal from between the horns and index-piece (fig 8).

In the succeeding operation a cylindrical cutter, *d*, removes the superfluous metal from either side of the index-piece, thus leaving only a narrow straight tongue of metal below the horns, which must be formed of the required shape (fig. 8).

The only point that presents any difficulty is the formation of the lever notch or the rectangular opening in the fork, for it is essential to first remove the strip that projects between the two horns (*b*, fig 7), and subsequently to cut the notch; and this, too, without damaging the index-piece.

The operation can be performed with the aid of a small mushroom-headed chamfering cutter, whose diameter is equal to the width of the lever notch.

The work may be simplified by first drilling a hole, where the notch will ultimately be, that is of exactly the required diameter but must not reach the index-piece. As the width of this hole will be somewhat greater than that of the projecting strip, this latter will fall away, and the diameter of the semi-circular space that is left will fix the position and the width of the small mushroom-headed chamfering cutter used for forming the notch.

It could also be accomplished by employing a cylindrical cutter similar to that used for cutting the U-spaces of cylinder

escape-wheels. If held in a vertical position, a backward and forward movement should be communicated to it; if horizontal, it should be moved from above downwards.

768.—Levers have also been made on which a small stud, of the same piece of metal, was left. The index-piece was screwed in this or held by friction.

A specimen of this form is represented in fig 9, plate VII.

The metal between the horns can then be removed in the ordinary way, that is with a simple cutter as shown at *a*, fig 7 of this plate.

Pivots and Pivot-holes.

769.—All the pivots of the escapement should have jewelled pivot-holes. The English as a rule do not employ them for the pivots of the pallet-staff, and even those of the pivoted detents of pocket chronometers are usually without them. These two mobiles can without detriment be made to work in brass pivot-holes, since experience has shown that oil will remain fluid on the pivots for a longer period when the movement takes place alternately in opposite directions; but in French watches, where the motive force is considerably less than in those made in England, it is essential that the holes be jewelled, for this is recognized as being one of the means that should be adopted for reducing the resistance opposed by the inertia of the pallets, the friction and the oil, to any rapid movement.

For a similar reason endstones should be applied to all the pivot-holes.

All the jewels employed in the escapement should be Oriental rubies or sapphires. We have elsewhere explained the reason for this.

770.—The best form for the pivots is shown in fig. 2, plate VIII. As we have previously explained, this allows of their being made somewhat finer than other forms, while at the same time securing a greater degree of solidity.

It is essential, in order that accurate timekeeping may be ensured, that the pivots be truly cylindrical; the watchmaker will do well to remember this fact and to avoid the use of pivot files; turning them with the graver until so fine that they can be finished with two burnishers, one rough and one smooth.

771.—The play of the balance-pivots in their holes should be a mean between that recommended for the cylinder escapement (**415**) and for the duplex (**555**). The play of the duplex

pivots would not suffice for the balance of a lever watch, and that allowed for the cylinder pivots would be too much. When the holes are too large the timing will be unsatisfactory, since the action of the fork or of the pallets will be variable or will give rise to loss of time in each change of position; but when too close fitting, the escapement will be very sensitive to the thickening of oil. On the whole, less inconvenience is experienced when the holes are slightly large than when small, providing the several actions are certain.

See articles **415** to **417** and **555**.

Practical details on the Balance-Spring.

772.—With a detached escapement, in which the vibrations are of considerable extent, the balance-spring should be long, and the workman will do well to employ one that is isochronous or approximately so.

Isochronism does not depend upon the number of coils but on the length and the general character of the metallic band.

In a chronometer, where the diameter of the cylindrical balance-spring is about a third that of the balance, isochronism is usually found to exist at about the eighth coil, whereas in the ordinary watches of commerce it is met with at about the twelfth coil.

It must be clearly understood that this does not imply that every cylindrical spring of 8 turns or flat spring of 12 turns is isochronal; and it is of importance here to remind the reader of the observations previously made (**652**) as to the influence the adoption of any one length of spring in preference to any other has, in the course of time, on the going of the escapement.

773.—As the cylindrical spring cannot be employed in ordinary watches, a curved balance-spring, known as the Breguet spring, is generally preferred. This form allows of its being made of considerable length and the coils sufficiently far apart to avoid all risk of contact between any two succeeding ones, or even of several of them when the arc of vibration is of unusual extent. It, moreover, secures very great regularity in the movement of the coils, which open and close in symmetry without moving towards one side, as is always the case with common springs.

The diameter of the balance-spring collet should be rather small.

The spring must not be strained or distorted at any point of its length, especially where the attachments are made to the stud and collet. Further details relating to this subject, which

are applicable to all detached escapements, will be given in a collected form in the article on the *Isochronal Spring*, and the reader will do well to select those passages in the discussions of the balance-springs of the cylinder and duplex escapements that are applicable to the present case.

Practical details on the Balance.

774.—We could only repeat here portions of the articles **422**, **558**, etc.; we therefore must refer to them.

The lever escapement has given satisfactory results when an ordinary balance of suitable dimensions was employed; but its regularity is greatly improved by the application of a compensation balance.

775.—The balance of the lever is usually a little heavier than that of a cylinder escapement; for the unlocking requires a considerable effort and this can only be efficiently applied by a rather heavy balance (**423** and **440**).

776.—The average diameter is twice that of the escape-wheel, but in the best watches, where the motive force is abundant and the escape-wheel rather less than in ordinary watches, the mean diameter of the balance may be two and a half times that of the wheel (**559**).

777.—The escapement in a gentleman's watch of the ordinary size usually has 18,000 vibrations per hour (**424**).

In English watches, which, as a rule, only beat 16,200 vibrations, the diameter of the balance is generally about two and a half times that of the wheel.

These data, as well as those given in the last article, are only mean values obtained entirely by empirical methods, and they must be made more complete and accurate by the considerations contained in the chapters relating to the Balance-Spring, the Balance and Timing (Part III.).

Under the second of these heads will be found all the details that are required in order to enable the workman to make a compensation balance. We will, therefore, only observe, in anticipation, that this addition demands most perfect workmanship, and the bi-metallic rim must not be cut until the whole is completed. With a badly made compensation balance the timing will be uncertain, and its variations are even greater than those met with with a well-made ordinary balance.

Verification of the Escapement in the Depthing Tool.

778.—Place the wheel and pallets between the centres of

the depthing-tool, the pallet-staff carrying a pointer* for indicating its motion on a small graduated disc held by friction on one of the centres, as shown at fig. 30, page 230; the two arms of the tool are then gently brought together until, on turning the wheel, the pallets are caused to traverse a total lifting arc of 10°.

The central line on the graduated sector should accurately bisect this angle.

Guiding the wheel and pallets with two fingers, it will be observed that this total lift of 10° divides itself into two unequal and distinct portions. The *first* comprises the period during which the point of the tooth traverses the locking face, terminating when it reaches the delivery edge of the impulse plane; the *second* is occupied by the passage of the tooth in contact with this edge. But since these two actions succeed one another, and, so to speak, together complete the angle through which the lever is moved, it is manifest that any increase of one involves a decrease of the other; thus, if the locking occupies 3°, the half-lift is 7°, and must, therefore, commence 2° before the line of centres, a fact which will be indicated by the index as it traverses the sector.

If the locking measure 2°, the half-lift will commence 3° in advance of this line, and so on.

When the half-lift commences too near the line of centres, this arises from the fact that the inclines of either the pallets or teeth are too low. On the other hand, if it commence far from that line, the inclines on one or the other must be too high.

In the first case the extent of locking will be excessive, while it will be deficient in the second.

The draw of both pallets must be verified. The movement of the index over the sector will indicate whether they are sufficiently near equality. The direction of pressure of the tooth should be *below* the pallet-centre for the engaging pallet and *above* for the disengaging pallet (**699**).

When much resistance is offered during the draw, the locking should only measure 1·5° or 2°; when the opposing force is slight, the locking should extend to 3° or 4°. It must be left to the experience and skill of the maker to decide upon the exact amount of this locking, which should in all cases be made thoroughly *safe*, but, at the same time, as small as practicable.

* The index formed of thin brass wire, softened so that it may be bent as required, is carried by a small screw ferrule fitting the pallet-staff (or the arbor), and motion is imparted to the escapement by a gentle pressure on this ferrule.

When the height of the impulse planes of the pallets is determined, the amount of locking may be reduced as required, by making the extremities of the teeth more and more inclined.

We know that the impulse faces of the pallets are usually inclined to such an extent as to secure an angular movement of the lever of 5°. This can easily be verified with pointed teeth, and, although apparently more complicated, it is in reality just as easy to do so when they are clubbed. The point *i* of the tooth (fig. 49, page 427) is caused to traverse the entire length of the impulse plane, but must not be allowed to pass beyond it, that is to say the corner *i* is arrested at *s*, as indicated in the figure. The pallet has by this means been impelled through the interval comprised between the two arcs *d*, *s*, or the height of the impulse plane, and the position of the pointer indicates the measure of this height in degrees. Were the tooth to continue its advance, it would impel the pallet through a total height measured by that of its own incline plus that of the tooth itself.

If the internal and external drops are found to be too short it is due to the fact that the teeth or pallets are too broad. On the other hand, drops that are excessive but equal inside and out indicate that the pallets or teeth are too narrow.

Considerable drop externally and none at all within prove the pallets to be too much closed or else that they have been left too long; excessive drop within and a deficiency without show that the opening of the pallets is too great, or that they are too short.

These faults may also arise from one of the pallets not being correctly proportioned.

After making all the requisite corrections and completing the verification, describe on the plate of the watch, from the centre of the escape-wheel, the circular arc on which the pallet-staff will be subsequently pitched.

The wheel is now removed from the depthing tool and replaced by the balance-staff. This latter carries its roller and an index, for observing the *total lift* on a second graduated sector (fig. 30, page 230), held by friction on one of the centres that carry the balance-staff.

Bring the ruby-pin into action in the lever notch and turn the two sectors until each index travels to the same distance on either side of its central line during the backward and the forward movement of the ruby-pin.

While guiding this latter, move the lever until the pallet-

staff index has travelled just beyond the 5° measured from the central line, and, holding the lever stationary in this position, make sure that the ruby-pin can enter and escape from the lever notch without coming in contact with the horns, and that, on entering, it engages safely against the side of this notch.

The interval of safety allowed between the ruby-pin and the horn should be slightly greater than that left between the roller and guard-pin.

There must be a sufficient freedom left between this guard-pin and the passing hollow.

The ruby-pin being brought within the notch, move the lever backwards to the same distance on the opposite side of the central line, and again verify the freedom as above explained.

Should the total lift (between 45° and 55°, **701**) be insufficient, it arises from the ruby-pin being placed too far from the centre of the balance-staff; the lever will thus be relatively too short.

When, on the other hand, this total lift is excessive, the ruby-pin will be found to be too near to the centre of rotation, so that the fork-arm is too long in proportion (**718**).

The several points must be corrected in accordance with the special requirements of the escapement under consideration. When it has been verified in each particular, describe an arc of a circle, with the points of the depthing-tool, on the plate of the watch, from the centre of the balance; this arc will intersect the circle already drawn from the planting point of the escape-wheel, and the point of intersection gives the centre for pitching the pallet-staff.

Ordinary Verification of the Escapement when in Position.

779.—If the plate of the watch is not gilt nor the holes jewelled, the escapement can be verified in position, as done on the calliper (**762**).

The reference marks, termed *lifting points*, having been made in convenient positions, the mainspring wound up and the balance checked by a piece of paper, the balance is set in motion, and the total lift, the height of the inclines, length of the locking faces, the draws and the drops are severally tested. Some practice in verifying on the depthing tool will make the workman able to judge with considerable accuracy by the eye alone, and to estimate these several proportions at once.

Arresting the balance in the middle of its path, make certain that the ruby-pin has enough freedom in the notch.

Cause the balance to perform a complete rotation on either side of the neutral position and shake the lever between its banking and the roller, in order to be sure that the interval of safety is sufficient and that the lever cannot overbank. Throughout this operation, except when within the passing hollow, the guard-pin should touch the roller when pressed towards it; were it to fall short in the least, this would show that the ruby-pin rubs against the horns.

When the shake is of too great extent, owing to the roller being too small or the banking too far out, there is some danger, in the latter case, lest the ruby-pin should strike the base of the horn or the horn itself; and, in the first case, when the guard-pin is pressed against the roller, one observes the tooth to leave its locking position and fall on to the edge of the impulse face, or even to advance a little along this incline. This fault is very serious and renders a change of the table-roller essential. (We have seen that such a fault may also arise from the inclines being too high or from lockings of insufficient length.)

Ascertain that the locking with each individual tooth is safe, and that, when a pallet arm passes into a space, a slight amount of freedom is left between it and the heel of the preceding tooth.

Holding the lever alternately against one or the other banking, move the tooth that has just escaped backwards, so as to make sure that there is sufficient freedom between the extremity of the pallet arm and the heel of that tooth.

It is always advisable in a lever watch to cut a hole below the escapement in such a position that the eye can observe the action of the ruby-pin; for there is some difficulty experienced in making certain that the ruby-pin strikes directly against the side of the lever notch and does not come in contact with the horns. The shaking of the lever suffices to prove whether they come in contact when the lever is against its banking, but it is impossible to ascertain by such a method whether they do so when the guard-pin touches the roller, in other words when a shake has brought the lever away from its banking. It will be evident that this friction at two points is very detrimental.

By observing with a powerful eyeglass through the passing hollow, while guiding the balance with the finger, it is nearly always an easy matter to ascertain whether the ruby-pin engages well (on its entrance) with the side of the lever notch;

but it is not so simple to assure oneself that it does not come in contact with the horns. Polishing rouge, applied in a thin layer, which must not be too liquid, first to the side of the ruby-pin and then to its face, will suffice in the absence of any other means for detecting such a fault; but very great care and delicacy are required to avoid being deceived.

This final verification can be easily accomplished with the aid of a very simple little appliance which any watchmaker can make for himself in accordance with the following description.

Ruby-pin Gauge.

780.—This consists of a round steel rod (*a b*, fig. 1, plate IX.) fixed in a handle and terminating in a pivot in which a shallow conical hole (*a*) is drilled; on this rod a tube c, having a projecting piece at its lower extremity, is held by friction, being movable by the pressure of the nail. The projecting piece gives support to a small bent bar *d*, and this can be caused to move in and out radially by a screw passing into the projection. The motion is constrained to take place radially by the pin *n* and the slide *i j*, having two arms which pass on either side of the tube.

The lower horizontal arm of *d* receives a pin that can be changed as required, being clamped by means of a screw.

The reader has doubtless already observed the use of this appliance.

After having secured in *s* a ruby-pin of similar dimensions to that of the escapement under examination, the maker places the lower pivot of the balance in the hollow *a*, and, when the two pivots are thus held end to end, it is an easy matter to set the pin of the tool by means of the screw, so that it exactly corresponds with that of the roller.

The remainder of the operation will be understood from an inspection of the figure: the pivot *a* being held in the oilcup of the balance jewel, and the rod *a b* held perpendicular to the plate of the watch, the lever is caused to move by turning this rod just as though it carried the ruby-pin.

The rod *a b* must not be too short: for if it were so it would be very difficult to ensure the perpendicularity of the tool, which must be light and accurately made.

Planting the Escapement.

781.—The only difficulty connected with the planting of the escapement consists in the exact determination of the positions of the shoulders or the extremities of the pivots. This can be

effected satisfactorily by complying with the directions contained in articles **473**, **474**, etc.

Timing in Position.

782.—For ordinary watches the methods explained in article **431** will suffice; and for rapid timing see **432** and the succeeding articles.

For the regulating of the best class of watches see the chapter devoted to the subject of springing and timing in the Third Portion of this work.

CHAPTER IV.

CAUSES OF STOPPAGE AND VARIATION IN THE LEVER ESCAPEMENT.

To Examine the Escapement.

783.—Very great care is essential in the examination of a lever escapement, and the watchmaker should always proceed on some definite method, for, in addition to the difficulty of detecting certain faults, very different circumstances may give rise to identical results. For example:

A tooth falls on to the impulse face of the pallet instead of on the locking face, a circumstance which may arise from (1) the wheel having some long and some short teeth; (2) one or even both pallet-arms too short; (3) excessive inclination of the impulse planes; (4) distance between centres too great; (5) the combination of two or more of these causes.

In the first instance the action of the escapement in all positions should be studied by the ear. The sound at the lift will not be one sharp blow, for it comprises two nearly instantaneous impacts. Notwithstanding this it is easy with a little care and practice to ascertain whether the action is satisfactory; any grating proves the existence of some fault, either in the escapement or the balance-spring, always of course assuming that the mechanism has not reached such a state as to require cleaning; for an escapement that is dirty or in which the oil has thickened never gives out a clear sound.

In a word, the watchmaker must never interfere with any part until he has satisfied himself as to what is at fault; for otherwise he is liable to mutilate the escapement and thus to be

obliged to alter it throughout, a contingency that can only be honestly avoided by a workman that is absolutely certain of what he is about.

Setting.

784.—A setting arises from one of the following causes:

(1) Motive force insufficient; a fault which is due either to defective depths or to a weak mainspring.

As we have already observed, without a sufficient motive force the lever escapement can never secure good results.

(2) Acting surfaces of the pallets and teeth badly polished. This circumstance materially diminishes the extent of the arc of vibration.

(3) Excessive draw, whether on one or both pallets.

(4) Total lift of too great extent. As the action of the fork takes place at a continually increasing inclination to the direction of the ruby-pin, the force applied to the lever is more and more decomposed; and, as the lift becomes greater, the effective impulse on the ruby-pin diminishes, while its resistance is increased. With even 60° of lift, the resistance opposed is already considerable towards the conclusion.

(5) Inclines (whether on the pallets or teeth) of insufficient height. The tooth will butt against the locking face instead of engaging with the impulse face of the pallet.

Such a setting may occur only on one pallet and be due to the balance-spring being wrongly pinned in the stud.

(6) A badly polished lever notch, causing the action of the ruby-pin to be accompanied by very harsh friction.

(7) Lever notch too narrow, so that the ruby-pin is constrained in its action, a fault which may also arise from this pin being too large or set out of square.

(8) The entire escapement, the wheel, pallets, lever, roller and especially the balance being too heavily constructed; such a fault is unfortunately very commonly met with in the productions of several Swiss factories.

We have omitted from the above enumeration all such causes of setting as are due to the absence of oil where it is essential, its thickening, or the presence of oil in too great quantity.

Roller too large,—too small,—out of truth,—badly polished.

785.—When the roller is too large there will not be sufficient freedom between its edge and the safety-pin.

If it be too small, the lever will overbank and pass to the other side whenever it is caused to leave its banking by a shake; or, if the roller will not allow the lever to actually overbank, the contact of the safety-pin with the edge of the roller will take place so near to the line of centres that the friction produced will be very harsh, and may at times be sufficient to occasion the stoppage of the watch.

With a roller that is out of truth, the faults that characterize a large roller may be met with in certain points of its circumference, while at other points it has those of a small roller.

If the edge is left square or badly polished, occasional contacts will take place, and the friction that occurs will at times be sufficiently harsh to stop the watch.

Oil on the edge of the roller and therefore on the guard-pin will disarrange the timing.

Guard-pin or dart too long,—too short,—or out of upright.

786.—It often happens that the fault does not consist in the roller being too large or too small but in the dart being too long or too short; this corresponds in English watches to the guard-pin being set too far out or not sufficiently so.

When the dart is too long it must be shortened with care so as to maintain its angle accurately in the centre of the lever notch; for otherwise, when the lever is held against its bankings, the dart will be nearer to the roller on one side than on the other.

The angle and two faces of the dart must be highly polished and kept free from oil.

If a dart of such a form as F, fig. 2, plate VII., is too short and the roller cannot be remade, the point of the dart must be filed away and a fine firm pin fixed in its place near enough to the roller to allow no more than the requisite freedom.

Some watch-jobbers, after having softened the lever, make a cut by means of a screw-head file across the base of the dart so as to separate its apex from the lower portion through about three-quarters of the total thickness; then, by striking a screw-driver placed in the slit, they gently advance the summit of the dart. Such a practice cannot be recommended.

The dart must not be wide at its base; it should only touch the edge of the roller by its extreme point.

When of the form shown at *r*, fig. 2, plate VIII., it may be lengthened by being struck with a small punch shaped like a hammer-head.

Lever notch too narrow,—too wide.

787.—In article **784**—(7) the disadvantages of a narrow notch are indicated.

The period during which the fork, in transmitting the impulse, acts on the ruby-pin, becomes less as the difference between the size of the lever notch and that of the ruby-pin is increased; and the energy of the impact varies directly with the amount of play allowed to the pin in the notch.

A lever notch that is too wide sometimes also causes the corner of the passing hollow to press against the dart or guard-pin, a fault which must be avoided at all costs.

Too short an engagement of the fork and ruby-pin.

788.—This occurs when the impulse faces of the pallets are not of sufficient height, or when the ruby-pin has too much play in the lever notch: at times it results that the roller presses against the guard-pin towards the termination of the impulse; this has a very detrimental effect and renders timing impossible.

It is important to remember that the ruby-pin during the entire impulse, that is to say until the lever rests against the banking, should be pressed upon by the same side of the notch without interruption, and that it must not come in contact with the horn or the opposite side either during the impulse itself or on escaping from the notch.

Inclines that are too high,—or too low.

789.—When the inclines, whether on the pallets or teeth of the wheel, are too high, the lockings will be very slight. With inclines that are too low, the amount of locking is excessive and the impulse will be considerably diminished. (The *real* half-lift should never fall short of 5° or exceed 8° of angular movement of the lever.)

Total Lift too great,—or too small.

790.—The disadvantages of an excessive total lift have already been indicated (**784**—4).

A deficient total lift is occasioned by the distance between the centres being too great, the inclination of the impulse faces insufficient, or by the ruby-pin being set at too great a distance from its centre of rotation; it is just the reverse of a too great lift. If the angular movement of the lever is not deficient, the position of the ruby-pin must be altered, that is to say it must

be brought nearer to the centre of the balance-staff, and the balance itself re-pitched in a convenient position (**716**, etc.).

Too strong or too weak draw.

791.—An excessive draw, especially if there is a long acting locking face, renders necessary a very heavy balance and a considerable motive force.

With weak draw and a short locking face, the lever is easily brought away from the banking, more especially when the wearer rides on horseback.

As we have already seen, if the draw is strong, the amount of locking may be reduced (to 1·5° or 2°); with a slight draw, the locking must be increased (to 3° or 4°); but it must be remembered that as the extent of locking is increased the resistance to unlocking becomes greater.

A too deep pitching of the teeth and locking faces and too strong draw always give rise to irregularity in the going of the escapement.

Excessive drops.

792.—The precise extent of the drop varies with the recoil and the length of the locking faces; it is, then, only by reducing these to within the narrowest possible limits that we can make the extent of the drops a minimum.

Pallets that are too much open or closed and pallet-stones or teeth too narrow, increase the drop, and, adding to the amount that is unavoidable on account of the draw, give rise to the very great drops that are to be observed in many escapements.

When the drop on to the engaging pallet is excessive and zero on the disengaging pallet, it proves that the pallets are too much closed; in the converse case the pallets are too open, etc., (**778**).

If the drops are excessive but of equal extent on the two pallets, it is evidence that the clubbed teeth of the wheel or the pallets are too narrow.

Excessive drop on only one side sometimes indicates that one of the pallet arms is too short or too narrow.

Unequal drops can only exist when a wheel is not perfectly round, or when its division has been carelessly effected.

Inequality in the extent of the lockings on either pallet arm has the same origin.

Too little Locking,—or None at all.

793.—If the tooth falls on the entrance edge of the incline or on the incline itself, the watch, although it may not stop, cannot be timed and the wear is rapid.

The escapement must be examined in order to ascertain whether it will be most advantageous to re-pitch the pallets or replace the pallet-stones by others with inclines of a different height, or merely to loosen the stones and advance them slightly; but the maker must first carefully note how the proposed change will modify the extent of angular movement of the lever.

If the stones are moved a little forward, their form can be altered as required, the inclination may be increased or diminished, and the faces subsequently re-polished with diamond powder.

Inner faces of the horns badly formed.

794.—Escapements are sometimes met with in which the inner curve of the horn has been struck from the centre of the balance. Such a form, adopted by the makers so as to enable them to polish both faces by means of a disc at one operation, is very objectionable. It leaves a prominent corner at *v* (fig. 3, plate VIII.) which may be touched by the ruby-pin and the fault may not be easy of detection. Contact between this angle and the front of the ruby-pin might suddenly stop the watch.

The curve of the horns should, as we have already observed, be such that, when the fork is in the position indicated by fig. 3, plate VIII., the edge is very nearly parallel to the circumference traced out by the front of the ruby-pin.

Some makers prefer it to be exactly parallel and of the same radius (it *must* be a trifle greater); but it is then necessary to round off the extremity of the horn, so as to prevent the ruby-pin striking against this point in consequence of any flaw in the roller.

Summary of other causes of Stoppage and Variation.

795.—The other causes of stoppage and variation in addition to those already enumerated either in this chapter or the three that precede it are:

PALLETS OR LEVER TOUCHING THE HOLLOW or having so little freedom that the oil of the pallet-stones spreads over it.

TOO SHORT PALLET-STAFF: the oil from the pivot spreads between the pallets and the bar.

TEETH TOUCHING THE MIDDLE OF THE PALLETS: where they deposit some of their oil.

PALLETS AND LEVER THAT ARE NOT IN EQUIPOISE ON THEIR AXIS.

TEETH BADLY POLISHED and whose points are rough.

RUBY-PIN TOO LONG so that it rubs against the hollow in the plate; or TOO SHORT, causing it to occasionally engage with its point on the flat of the horn in consequence of the bending of the lever or endshake of the pivots.

RUBY-PIN CARELESSLY CEMENTED and projecting forward so as to come in contact with the horns; or inclined to one side, when the fork will force the passing hollow more towards one side than the other.

RUBY-PIN AND PALLET-STONES NOT FIRMLY CEMENTED.

ROLLER SET OUT OF FLAT ON ITS STAFF, a fault that is often met with when a shoulder has not been left at the centre.

A BAD COMPENSATION BALANCE whose arms disturb the equilibrium on a change of temperature, or come in contact with any part on opening outwards.

SEVERAL COILS OF THE BALANCE-SPRING COMING IN CONTACT IN THE LONG ARCS.

THE BALANCE-SPRING NOT OF UNIFORM STRENGTH, or one that has become INELASTIC or been DISTORTED in long vibrations.

LEVER NOT FIRMLY FIXED ON THE PALLETS.

LEVER HOLLOW TOO SMALL; the thickening of oil, or a particle of dust interposed between the lever and the side of the hollow against which it banks, will suffice to stop the watch or interfere with its rate.

It is best to make the lever bank against two pins.

PASSING HOLLOW TOO LARGE OR SMALL. In the latter case the roller touches or grazes past the guard-pin when the ruby-pin enters the notch; when too large, it may happen that a shake displaces the lever before the ruby-pin has entered the notch; there will then be contact between this pin and the inner face of a horn.

HORNS TOO WEAK AT THEIR BASE: they are liable to be strained by a sudden blow when the balance banks against them.

LEVER LONG AND FLEXIBLE with its bankings improperly placed. It bends at every banking of moderate energy. This fault is of some importance when the banking takes place against the axis of the escape-wheel (**683**).

OIL passing down the axis of the escape-wheel and causing the two arms of the lever to adhere (x, x′, fig. 1, plate VIII.).

MAGNETISM, whether acting between the wheel and the steel pallet-arms, the two arms of the lever and the escape-wheel axis, or the steel fork and a pin of the same metal working in the notch.

GUARD PIN BENT in cleaning or by banking.

RIM OF THE BALANCE TOUCHING THE CHAIN in English three-quarter plate watches.

CHAIN RUBBING against the top plate, the hollowing of the cap, the potence, and occasionally against the case.

BALANCE TOUCHING the cap, in full-plate watches, etc., etc.

The reader should select from the lists of the causes of stoppage and variation of the cylinder and duplex escapement, etc., those that are applicable to the lever.

NOTES

ON SOME DIFFERENT FORMS OF THE LEVER ESCAPEMENT.

Lever Escapement of the most simple construction.

796.—The exact form of this escapement, which was proposed by Perron in 1798, has been described in the *Revue Chronométrique*.

The impulse planes are on the teeth of the escape-wheel (fig. 2, plate IX.); the lever and pallets are formed out of a single piece of brass, from the surface of which two small pins, *b* and *g*, project, and these take the place of the ordinary pallets. The extremity of the lever is hollowed out in the middle and terminates in two small horns whose external faces curve inward; so that these convex faces rest against the edge of the roller and prevent any motion of the lever except during the period of lift. This is effected by means of the ruby-pallet *a* cemented in the roller.

The teeth of the wheel must be sufficiently cut away at the back, and the front faces must be sloped so as to occasion such an amount of draw as shall ensure the steadiness of the lever against its banking. The pins fine rather than thick; and half of their thickness may be removed, thus leaving their horizontal section semicircular. If very perfect action is desired their faces should act like the cylinder edges in the horizontal escapement.

In conclusion, the simplicity of this arrangement is more apparent than real, for it requires very great care in its construction, since otherwise its accuracy cannot be relied on; those who have proposed to replace the pins by small rollers (Perron had already suggested their use in 1798) merely introduce complications without any improvement.

English Lever Escapement.

797.—In the best class of watches, the escapement known in France and Switzerland as the English Lever mainly differs from those represented in plates VII. and VIII. in having the escape-wheel teeth pointed. The escapement is usually right-angled, and the distance between the centres of the lever and balance generally equals the chord contained between the points of five teeth, with a 15-toothed wheel. An empirical rule places the ruby-pin at one fourth this distance, so that the lever will slightly exceed three fourths.

Most English watches are made with covered pallets. They are separate from the lever, being attached to either surface. Fig. 4, plate IX. represents such an arrangement; and various forms may be adopted for the fork and roller action.

In one a pallet or finger is cut in the roller (A, fig. 3, plate IX.) in place of the ordinary ruby-pin; the effect is identical with that of the system explained in article **796** and has the same objection, namely that it brings the point of contact of the guard-pin and roller too near to the line of centres.

In another form the ruby-pin is replaced by two pins *a*, *n* (fig. 4, plate IX.); and a guard-pin, *b*, fixed in the fork in a position corresponding to the point of the dart, answers the same purpose.

This system facilitates the manufacture of the escapement, but there is danger of the pins being strained by a careless workman or by a violent banking against the back of the horns. If the pins are at all thick the roller must be thick also; or else the pins must be made with shoulders, carefully riveted, filed away on their front faces, etc., and this involves quite as much labour as the making and fixing of a good triangular ruby-pin, if indeed it does not require more.

One of the most successful modifications of the ordinary fork and roller action is that proposed by Savage of London, who replaced the ruby-pin by two fine pins, between which is a notch in the roller; the impulse is communicated by a pin,

carried by the lever, engaging in the notch, and this also serves the purpose of a guard-pin. The escapement, however, requires very perfect workmanship.

Lastly, the form shown at B, fig. 3, plate IX., is occasionally met with.

The extremity of the lever, instead of being formed into a notch, is merely a dart working in conjunction with a roller, and in the lever two pins are fixed perpendicular to its face which answer the purpose of the sides of the notch in the ordinary construction. A pallet, having a fish-tail form and with its extremity concentric with the axis, takes the place of the ruby-pin.

We shall not discuss these varied arrangements any further, nor others that may be occasionally met with; the reader should now be in a position to take an intelligent view of them.

Cole's Lever Escapements.

798.—Struck with the irregularities occasioned in watches by the violent banking of the balance against the back of the horns, J. F. Cole, of London, has experimented on various means for avoiding this effect and gives preference to the two following.

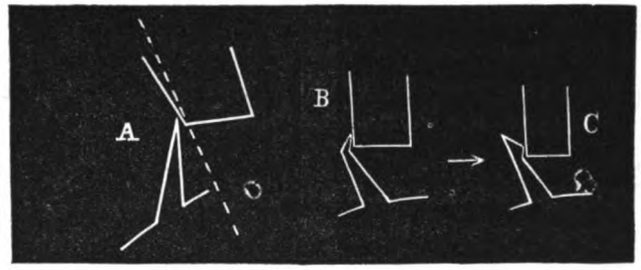

Fig. 51.

In the first, known as the *resilient* escapement (fig. 51, B for ratchet and C for clubbed teeth), the usual incline of the front face of the teeth (about 27°, B A D, fig. 3, plate VII.), instead of being continued to the base of the tooth is only made of the depth necessary for locking. The remaining portion of the face slopes in the opposite direction and thus prevents the movement of the pallet beyond the locking point, restoring it at once to this position whenever displaced by banking. There is therefore no necessity for banking pins and the force exerted is of an elastic nature and in no sense a blow.

In his other form of escapement, termed *repellent*, the locking faces are inclined outwards (A, fig. 51), thus causing a tendency of the pallets to unlock; the train, therefore, will run down on the removal of the balance. This tendency to fly out keeps the extremity of the lever, which instead of being a fork is a thin pointed edge, always in contact with the (jewelled) roller, and thus the escapement becomes closely analogous to the duplex. A notch in the roller allows of the release of a tooth, when the impulse is communicated; it will be observed that this form also renders banking pins unnecessary.

Note on pallets with circular locking faces.

799.—In the concluding paragraph of article **631**, it is stated that the steadiness of the pallets can be maintained equally well whether the locking faces be curved or straight. We should have added that this is only possible, in the case of faces concentric with the axis, if the arms are of very unequal length.

Lever watches may occasionally be met with provided with circular steel pallets, such as were originally employed. If well made they go fairly satisfactorily so long as they remain in one place but for the most part become irregular when carried.

We have always found it possible to correct this fault, providing it was the only one, by changing the direction of pressure on the locking face; but then a slight draw was occasioned and the wear of the locking faces became rather more rapid.

Our theory of detached escapements establishes clearly the necessity of maintaining the steadiness of the pallets, and experience confirms its conclusions. It must be carefully borne in mind that the irregular displacement of the pallets is mainly due, not to circular motion round their axis, which indeed occurs very seldom, but to the effect of a shake and the reciprocal action between moving bodies.

We shall leave it to the reader to draw his own conclusions from the following facts: escapements with circular pallets fitted with rubies are found to vary in their rate so long as the oil remains fairly fluid, whereas when the pallets are made of steel these irregularities become less. This circumstance must be attributed to the fact that the tooth holds more firmly against the locking face of the pallet, when the polish has been removed from its surface by the continued action of the teeth.

DETENT OR CHRONOMETER ESCAPEMENT.

CHAPTER I.

Preliminary.

800.—The detent to which this escapement owes its name may be made of two principal forms and these constitute two varieties of the escapement.

In the one the detent, termed by the Swiss makers *bascule*, is mounted on a pivoted axis. The arm of this detent against which the wheel locks is brought back to its position of rest, whenever it has been displaced, by a straight spring fixed to the plate, or, more usually, by a flat spiral spring attached to its axis.

In the second arrangement the detent and spring form one piece; and this spring, which is merely a prolongation of the detent, terminates at the end farthest from the balance in a small block which serves as a means of fixing it to the plate. The elasticity of the spring or thin portion of the detent renders it possible to unlock the wheel by drawing the locking arm aside, and brings this arm back to its position of rest whenever it is so drawn aside.

The first of these is known as the *pivoted detent* escapement and among Swiss makers as the escapement *à bascule*.

The second is termed the *spring detent*.

The first detached detent escapement was invented and made by Pierre Le Roy (**589**).

F. Berthoud first employed the spring detent and he attached the small spring, often known at the present day as the *gold spring*, to the roller on the balance-staff.

The pivoted detent escapement has been constructed in France chiefly by L. Berthoud and after him by Motel. These two makers brought the workmanship to a high state of perfection and obtained excellent results.

J. Arnold adopted the spring detent of Berthoud in a modified form, transferring the unlocking spring from the roller to the detent itself and altering its shape.

Earnshaw subsequently proposed a different form of escape-wheel from that of Arnold; he made them flat instead of having teeth projecting above their surface; he also changed the direc-

tion of the pressure during locking. These modifications in conjunction with certain others, suggested mainly by Breguet, have resulted in the spring detent escapement as now constructed.

The mode of action of the chronometer escapement is simple, but it does not admit of any error in the application of its principles nor any inferior workmanship.

It absolutely requires an isochronal balance-spring and a compensation balance. It must be carefully treated by its owner and only works satisfactorily in veritable chronometers intended for scientific observations; it should never be employed in ordinary watches.

The remarkable regularity, mainly due to it, that is observed in the chronometers employed by astronomers, naval and scientific men, and constructed chiefly by English and French makers, has led the manufacturers of ordinary watches to fancy that they would secure more accurate timing by the mere employment, in their best watches, of more or less accurate copies of detent escapements. Their attempts have always turned out to be failures.

The higher class of watchwork cannot be attempted with any chance of success except by makers who, besides being skilful, possess a sufficient amount of scientific knowledge. Without this there can be no progress, and, although we may occasionally meet with good imitators, there are no real artists, much less originators.

A mere trading watchmaker should never offer a chronometer or watch for sale as a thoroughly reliable timekeeper unless it is by a maker of proved ability, well-known for the success of his marine timekeepers or for watches supplied to scientific men who are accurate observers. Let him remember that the name, however well it be engraved on the plate of a watch, cannot replace workmanship and make its rate reliable; and that for everyday use the detent escapement, even if perfectly made, will never be successful except in the most careful hands.

Denomination of the several parts.

801.—This escapement consists of:

(1) A flat wheel with pointed teeth. It is shown at A, fig. 5, plate IX.;

(2) A locking detent against which the wheel rests. This

is termed a *spring detent* (*p* B C, figs. 5 and 6) when fixed to the plate by a small foot, and made thin at C so that it bends about that point; and it is a *pivoted detent* when supported on an axis, as D F and M *r* in figs. 10 and 11 respectively. The spring which restores the pivoted detent to its position of rest is known as the *recovering spring*.

A small cylindrical ruby, half of which is cut away at the upper end, projects at right angles to the detent at the point B, figs. 5 and 6. The tooth of the wheel, when locked, presses against the flat face of this hemi-cylinder; hence it is termed the *locking stone* or simply *locking*.

The light spring *m n*, figs. 5 and 6, also fixed to the detent, is known as the *gold spring*, *unlocking spring*, or *auxiliary spring*. This presses against a pin *p* projecting from the detent body, so that it can be inclined from *p* towards *n*; but it cannot be deflected from *n* towards *p* without the detent moving with it;

(3) A balance carrying on its staff a steel disc D in which is cemented a ruby *impulse pallet* J. This disc is known as the *impulse roller*.

On the same staff and below the roller a second or *discharging roller* E is held by friction and it carries a ruby *discharging* or *unlocking pallet e*.

The screw *f g*, whose head determines the resting position of the detent, is often termed the *banking screw*.

Action of the escapement.

802.—If the mainspring be wound up when the balance is at rest, with the balance-spring in its neutral position, no movement of the escapement will ensue; there will merely be a pressure exerted against the flat face of the locking stone.

But if the balance be made to vibrate, by giving a rotatory movement to the entire mechanism, say from left to right, the discharging pallet *e* (fig. 5, plate IX.) coming against the extremity *n* of the gold spring *m n*, forces it aside and continues moving beyond it.

On being brought back by the balance-spring, the pallet *e* again presses against *m n*, forcing it to move in the opposite direction; but, as this spring is now opposed by the pin *p* of the detent body, the detent is forced aside, bending at the part C, and it is thus released from the tooth *v*.

At this instant the wheel is free. As the pallet J is now

slightly in advance of the tooth *r*, this tooth engages with it and the pallet is driven forward through a certain lifting angle.

The lift terminates with the tooth *s* falling against the locking stone B of the detent, which is brought back by its own elasticity, before the end of the lift, to the position it occupied before the impact of the discharging pallet.

The balance has thus received the impulse and performs the supplementary arc. The return vibration is *dumb*, since the only action that occurs during that period is the slight displacement, by the discharging pallet, of the gold spring, which yields so as to allow of its passage; the balance on again returning effects the next unlocking, receives a fresh impulse, and so on.

It will be seen that this escapement bears some analogy to the duplex, as the balance only receives an impulse at every second vibration.

It is liable to set; in other words it is unable to start itself from a position of rest when the mainspring is wound up. This necessitates that the balance be set in motion by a direct impulse, due to a shake or twist given to the entire mechanism.

PROPORTIONS RECOMMENDED BY VARIOUS AUTHORITIES.

SPRING DETENT.

803.—ARNOLD*.—Fig. 7, plate IX., represents Arnold's escapement as he made it for his best chronometers.

The teeth project above the flat of the escape-wheel and act by their curved sides against an impulse pallet directed towards the centre of the balance-staff. The pressure against the locking stone tends to elongate the detent; in other words it is directed from the detent foot *b* towards the extremity *a*.

The unlocking of the tooth is effected by drawing the detent *towards* the escape-wheel axis, so that the released tooth passes behind the locking stone.

804.—EARNSHAW† replaced Arnold's wheel by one that was flat with inclined pointed teeth, fig. 8, plate IX. He arranged

* Two watchmakers of this name have been celebrated in England, where the second, the son, followed up the successes of the first.

J. Arnold's chronometers were acknowledged to be superior to those of Mudge, and in 1790 they secured him a reward of £1320. The younger Arnold obtained in 1805 a further sum of £1680, making a total of £3000.

† The modifications that Earnshaw introduced in the detent escapement and the excellent rates of his chronometers secured for him an equal Government recompense in 1805.

Without wishing to detract from the merits of these makers, who are justly celebrated, we may be permitted to point out that, as early as 1784, L. Berthoud in France

that it should rotate in the opposite direction, and thus it results that the pressure of the locking tends to force the detent towards its foot. It is driven *from* the wheel at the unlocking.

The direction of the impulse pallet passed through the middle point of a radius of the roller.

805.—TAVAN. — This author refers to three different forms of escapement as Arnold's of which one was certainly not designed by that maker.

The principles of construction adopted by Tavan and inserted after his descriptions may be summed up as follows.

"There are no fixed proportions to be observed in the unlocking pallet: if it is short, it must be pitched deeper with the auxiliary spring; if long, the pitch should be less deep.

"If it be required to ascertain the length of impulse pallet that will secure a lift of 60°, the best adapted to this escapement, the following is the method of procedure:—With a 15-tooth escape-wheel divide the distance between the centres of the wheel and balance into 21 parts and take 15 of these for the radius of the wheel and 6 for the length of impulse pallet."

This author, considering it highly important to secure great certainty in the lockings, places the resting point of the locking stone a little short of the tangential position when the detent is to the right of the wheel, and a little beyond it when to the left, so that the wheel always has a slight tendency to draw the detent towards itself.

806.—JURGENSEN. — The following are the principal proportions of this escapement:

The radius of the impulse roller should equal the interval between the points of two teeth.

The face of the locking stone should be inclined so as to occasion a slight recoil of the escape-wheel at the unlocking. (In the first edition of his work he made the locking tangential.)

The unlocking spring can be directed towards the centre of the balance-staff. The action of the escapement is rendered more certain by making the unlocking pallet act slightly before the line of centres rather than beyond it, when on the point of unlocking the wheel.

The escape-wheel teeth should accomplish the lead as uniformly as possible, and (in Arnold's escapement), so far as

had introduced into his pivoted detent escapement most of the improvements that were so liberally rewarded in England.

is practicable, perpendicular to the impulse pallet. (In his first edition Jurgensen states that the curve of the teeth of Arnold's escape-wheel should approximate to an epicycloid.)

The position of the impulse pallet should be so related to that of the unlocking pallet that there is sufficient drop, between the unlocking and the fall of the tooth on the impulse pallet, to ensure the proper action of the escapement. This drop should be rather greater in pocket chronometers than in marine timekeepers that are supported in gimbals.

The detent should be displaced to such a distance by the unlocking pallet that it does not fall back against the banking screw until the liberated tooth has travelled to a distance of about a quarter the interval between two teeth beyond the locking stone.

The banking screw of the detent is to be placed as near as possible to the centre of percussion of the latter.

807.—MOINET.—The unlocking spring should point very nearly towards the centre of the roller, but so as to be struck *after* rather than before the line of centres, in the case of a spring detent. (This is the converse of what Jurgensen recommends.)

The banking screw and locking stone are set at about a quarter of the length of the detent from its extremity, this being approximately its centre of percussion.

All the other data given by Moinet are to be found in the previously published work of Jurgensen.

808.—A. BREGUET.—Fig. 9, plate IX. represents a plan of the spring detent as suggested by Breguet.

The locking is practically tangential to the escape-wheel, the tangent passing just in front of the centre of the balance-staff. The auxiliary spring is directed towards this centre or very approximately so, and reaches almost to a line joining this centre and the point of flexure of the detent. In very many English forms of the escapement its direction differs considerably from that here indicated.

Finally, the point of rest on the locking stone and the point at which the unlocking pallet is struck are so taken that the force effecting the unlocking and the resistance at the locking stone act on the two detent springs in such a manner that the detent itself is protected from the effects of torsion.

The banking is carried at the extremity of a small arm or bar parallel to the detent.

This system possesses certain advantages but it entails rather more labour than those previously considered. It might perhaps have been definitively adopted had it not been employed in conjunction with that objectionable arrangement which places the balance and roller at opposite ends of the staff. Irregularities in the rate, due in great part to this mechanical error, brought this form of spring detent into disrepute, although it was one of the best known at the time.

The Swiss escapement-makers have modified Breguet's detent, but unfortunately without understanding it. Thus they varied the position of the locking point and of the point at which the auxiliary spring is struck, and yet continued to make an aperture in the flexible portion of the detent; this converted what was a rational proceeding into an absolute contradiction.

809.—GANNERY.—He considered a certain amount of draw essential and made the lift about 45°.

The distance between the centre of movement of the detent and the extremity of the auxiliary (or gold) spring is to the radius of the escape-wheel as 2·33 is to 1, or 7/6ths of the diameter.

Its entire movement during the unlocking is 2° produced by a balance motion of about 22°.

When about to accomplish the unlocking the pallet should come into action a little before the line of centres; a double advantage is thus secured; the greater portion of its lifting action is employed to effect the unlocking, and the remainder of the arc becomes proportionately less; and this diminishes still further the already feeble resistance of the auxiliary spring.

PIVOTED DETENT.

810.—L. BERTHOUD, MOTEL.*—Fig. 10, plate IX. represents, as accurately as is possible in a drawing of such dimensions, the detent employed by L. Berthoud and subsequently by Motel.

The rest is tangential to the wheel. The direction of the

* Louis Berthoud (nephew of F. Berthoud) improved the calliper of French chronometers and constructed 150 of them. This is a great number for that day, especially when we consider their excellent quality. He died in 1813.

H. Motel was the pupil and successor of L. Berthoud; he acquired considerable reputation by the great number of chronometers he made; they were beautifully constructed and possessed remarkable rates. He died in 1859.

auxiliary spring is such that the action of the balance is practically equal on either side of the line of centres, *b c*, and it terminates close to this line. The spring was formerly made of steel and the pin against which it rested was of gold. The point of contact required to be oiled.

The necessity of applying this oil and of preventing it from passing to the unlocking pallet compelled them to form a projection in front of the pin, at the extremity of the auxiliary spring; this occasioned a slight flexure and a cutting action against the pin.

The surface of the impulse pallet where it engages with the tooth is so curved as to facilitate the engaging and the lifting action.

In the chronometers made by L. Berthoud and in very many of those by Motel, the inclination of the teeth is less than that indicated in fig. 10.

The banking of the detent consists of a pin fixed excentrically in the head of a screw which is firmly held in the plate.

A simple examination of the figure will make evident the mode of action of the recovering spring *r b*. Its effective length is about equal to the diameter of the wheel.

L. Berthoud's pivoted detent, amended by the use of a spiral recovering spring and a gold auxiliary spring at a less inclination to the detent itself, is an arrangement that bears comparison with the best escapements made.

If it has been less studied than the spring detent this fact must be mainly attributed to the practice of the majority of French makers of arranging the balances of their marine chronometers to beat 18,000 vibrations per hour; an excessive velocity which made every fault of construction or error in principle the more manifest; and it is further due to the high degree of finish and the manipulative details in which these makers took so much pride in their best chronometers, at the very time that the English-made instruments were characterized by marked simplicity.

811.—TAVAN.—In the memoir that describes the researches of this authority, all that refers to the detent escapement appears to be the result of a mere examination of chronometers by other makers and not from experiments made by Tavan himself.

In addition to two drawings of the spring detent, he gives

a very bad arrangement of the pivoted detent, and then concludes in favour of the latter in the following terms.

"The system that has the locking piece supported on a pivoted axis provided with a flat spiral spring to restore it to its locking position is preferable, because the resistance opposing the unlocking can be more easily modified by means of this spring. The locking action is more secure when the locking stone forms part of a rigid lever than when it is fixed to a flexible spring."

812.—MOINET.—"The pivoted detent is not thinned down to a spring and, instead of being provided with a foot, it has a fine axis at its extremity, the pivots rotating or rather oscillating in holes in the plate and a bar. A flat spiral spring of 3 or 4 turns and of sufficient strength is held by one end in an ordinary stud held friction-tight in the plate, while the centre is attached to a collet on the axis of the detent, precisely as in the case of the balance-spring of the ordinary balance.

"As the pivoted detent has been the less used, we are not possessed of comparative results that would suffice to decide us in our choice; the spiral spring, being the longer, would appear to be characterized by a less rapid change in the resistance; but the detent opposes a greater mass, and between these two effects, so delicate in the case of an escapement, it is impossible to decide except by special experiments, the nature of which appears, so far, to be but little known."

813.—M. HENRI ROBERT.—Figs. 11 & 12 of plate IX. represent, in plan and elevation, a pivoted detent escapement as made by M. H. Robert; he is, to the best of our knowledge, the only chronometer-maker, if we except M. L. Berthoud the younger, who has employed it in marine timekeepers.

"The front faces of the teeth are inclined in the direction $q\,o$, making an angle of 30° with the radius $n\,q$. This inclination is very generally adopted at the present day." The face of the impulse pallet l forms the same angle with the radius of the roller.

The small spring s is a strip of gold bent at right angles so as to form the foot s'; this foot has a slot cut in it, so that it can be passed under the head of the screw without the necessity of removing the latter.

The direction given by L. Berthoud to this spring was modified by M. Robert, who made it less inclined to the body of the detent and directed towards the centre of the balance-

staff. He also set the banking screw, *t*, nearer to the centre of motion.

Summary of this chapter.

814.—If the various forms of escapement that have been enumerated above be examined attentively, one is struck with the great differences in their several proportions, which seem to imply the absence of any precise basis founded on the laws of Mechanics.

When an art or industry first commences it cannot be otherwise: even science herself is forced to rely on carefully observed facts, on a mass of experimental data, before she can enunciate her laws; just as in mathematics it is impossible to determine the *unknown* terms except by the aid of quantities that are already known.

It is, then, not without some advantage that the reader can here examine, side by side, the principal proportions adopted by the above authorities.

Direction of the Impulse Plane.

815.—The face of the impulse pallet, against which the teeth of the wheel act, was directed towards the centre of the balance-staff by L. Berthoud, Arnold and Motel; and towards the middle point of a radius of the roller by Earnshaw and Breguet.

Direction of the Unlocking Spring.

816.—The small unlocking or auxiliary spring should be directed towards the centre of the balance-staff, or very nearly, according to all the authorities; but with this difference, that, whereas some consider the action should be equally divided on either side of the line of centres, others assert that there is an advantage in effecting it before this line rather than after, and a directly opposite opinion also has its supporters.

This difference in the mode of action might be caused by alteration of the direction given to the spring, and of the mode in which the small acting extremities are rounded.

The line joining the point of flexure and the free extremity of the auxiliary spring is almost parallel to the body of the detent in Arnold's and Earnshaw's escapements; on the other hand, it is inclined at a considerable angle in the escapements by L. Berthoud, Breguet and Motel.

Banking of the detent.

817.—The banking screw in the construction of Arnold

and of Earnshaw (*c* & *d*, figs. 7 & 8, plate IX.) may be regarded as a rigid obstacle. In that of Breguet it is a cam (*a*, fig. 9) carried on an arm whose only fault consists in its being too massive. Berthoud's arrangement (*a'*, fig. 10) is the best of those in use at his day.

Number of Vibrations and Lift.

818.—Earnshaw and Breguet made their marine chronometers to beat 14,400 vibrations per hour. L. Berthoud and Motel increased the number to 18,000 or even beyond that point.

A lifting angle of 60° was very generally adopted by the old makers.

Position of the Balance and Balance-spring.

819.—L. Berthoud, Arnold and Earnshaw placed the balance and impulse pallet near the middle of the balance-staff, and they thus conformed to the principles so clearly laid down by Pierre Le Roy. The balance-spring occupied the upper portion of the staff. Motel brought the lifting action too near to the pivot; but the system adopted by Breguet was, as regards this point, the most objectionable. He placed the balance at one end of its staff, the impulse roller at the other end and the balance-spring between them.

Escape-wheel and Impulse roller.

820.—Arnold and Earnshaw made their wheels with 12 teeth. Those of other makers had 15 teeth.

The size of impulse roller is variable, because it depends both on the number of teeth and the distance between the centres; and these quantities are not constant.

L. Berthoud and Breguet made smaller rollers than did the English makers.

All the early authorities employed very large wheels in comparison with those in use at the present day. The diameter has been gradually reduced and there is actually now some fear that makers will fall into an extreme in the opposite direction.

Proportion between the detent and wheel.

821.—Taking the radius of the wheel as a basis of comparison in each case, the following table gives the length of the detent, measured from the centre of flexure or from the centre of rotation, to the extremity of the auxiliary spring.

Arnold (the elder).—Spring detent	about	2.0
more recently	,,	3.0
L. Berthoud, Motel.—Pivoted detent	,,	1.1
Earnshaw.—Spring detent	,,	3.0
Breguet. ,, ...	,,	2.0
Gannery. ,, ...	,,	2.3
H. Robert.—Pivoted detent	,,	1.2
Arnold ,, ...	,,	1.4
according to Spring detent	,,	1.6
Tavan ,, ...	,,	2.0

On comparing these figures among themselves and examining the different forms of modern escapements, we at once notice two facts: (1) pivoted detents are much shorter than the others (measuring from the centre of motion to the end of the unlocking spring); and, (2) whereas the French makers have retained, very approximately, the same length of detent as was adopted by Breguet and have materially diminished the size of the wheel, the chronometers of English design are, as a rule, characterized by both shorter detents and smaller wheels than those of earlier construction.

OBSERVATION.

822.—The detent escapement, in its two forms, as made at the present day is a composite invention.

The principle of the free detent and the first escapement constructed on that principle are due to P. Le Roy.

F. Berthoud applied a flexible blade to the detent in place of the axis, and he attached to the roller a light unlocking spring bent at right angles.

Subsequently Arnold in England and L. Berthoud in France altered the form of this auxiliary spring, sometimes termed *deer's foot* spring, and attached it to the detent itself.

Breguet set the banking screw at the extremity of an arm, and very approximately determined the length of detent best suited to marine chronometers. Prior to him L. Berthoud had employed an arrangement for banking the detent that is preferable to the one in favour in England at the time.

Arnold's form of escape-wheel was abandoned in favour of the flat wheel with pointed teeth of L. Berthoud and Earnshaw.

The detent escapement, then, independently of any minor improvements, cannot be regarded as the invention of any one of the watchmakers we have quoted above, but it is the outcome of the collective labours and experiments of all these great authorities.

To attribute it solely to one of them and to describe it, for

example, as Arnold's escapement, which is still very often done in England and Switzerland, is a great injustice and utterly at variance with history.

CHAPTER II.

RULING PRINCIPLES IN THE CONSTRUCTION OF THE DETENT ESCAPEMENT.

General Considerations.

823.—The reader will do well to read again our observations in article **686**; and these we will complete and supplement by the few following remarks.

Chronometers by Arnold, Earnshaw, Berthoud, Breguet and Motel, as well as by their respective successors, have maintained excellent rates at sea. But the special arrangements adopted by these highly skilled watchmakers differed materially, for they employed long and short detents, large and small wheels, spring and pivoted detents; and in this we have evidence of the fact that success is not secured by the mere selection of this or that escapement or of a particular size for any part, but that it depends, as has been already demonstrated in our theory of escapements, on a number of proportions and a certain initial relation between the several elements of the mechanism; a relation which should be such that its modification by time (the thickening of oil, etc.) has the least possible effect on the rate of the mechanism, assuming it to be, in the first instance, properly timed.

We do not include among these disturbing causes the wear of the parts that is brought about by an unsuitable mechanical combination, the employment of inferior materials, careless workmanship, or the ignorance of workmen. When the mechanical execution is at fault everything becomes uncertain.

Thanks to the labours of our predecessors we are now able to realize an efficient arrangement, and to secure results that are somewhat better than those to which they attained, with almost absolute certainty, although we are still ignorant of the scientific

explanation of several points. But they have opened out the way and set the limits to it; it is for us to avoid straying from it.

THE TANGENTIAL ESCAPEMENT.

Is it possible for the centre of the balance and the locking to be on one and the same tangent?

824.—It is often regarded as an elementary rule in the workshops that, to secure the best results with this escapement, the locking point and the centre of the balance-staff should lie on a line tangential to the wheel.

This condition, assumed to be of primary importance, is open to the objection that it only deals with the question from one point of view, and it confines the problem within too narrow limits, since the data to be considered become too few.

The locking point can always be set tangential, but the position of the centre of the balance varies with the lifting angle.

825.—Let the circle *a r v* (fig. 4, plate X.) indicate the circumference of the escape-wheel, one tooth of which is held against the locking stone at *r*. The line *r c* is a tangent at this point *r*.

If the wheel has 15 teeth, three of these will occupy the positions *r*, *b'*, *b*; the balance must, then, be set at *d* to be on this tangent. The lifting angle *b d b'* will in round numbers be 70°.

With a 12-tooth wheel the three teeth will be at *r*, *a'*, *a*. The centre of the balance is then at *c*, and the lifting angle *a c a'* is 60°.

Under similar conditions a wheel with 10 teeth will give a lift of about 45°, and this is the amount most generally employed.

Both theory and experience have indicated the objections to an excessive lift (**97**), and we know that, in practice, it is possible to secure a sufficient extent of vibration of the balance by a lift of about 45°. Hence we shall do well to limit ourselves to this angle; but it will be evident the application of the so-called rule would compel us to employ a wheel of 10 teeth in order to do so.

Now, if the movement of the balance be adjusted, for example, to 14,400 vibrations in an hour, the 10-tooth wheel must move through an angular distance of greater extent in approximately the same period of time. It must, then, travel with a greater velocity than a wheel of 15 teeth; but, since an increased velocity corresponds to a much greater relative increase in the force producing it (**121**), it is evident that a greater motive force will be required.

We here have a new calliper of escapement to calculate and experiment upon; but a simple drawing at once shows that, ignoring the different velocities of the two wheels, when the one with 10 teeth is substituted for that with 15 teeth, it will involve longer acting surfaces, a larger detent and roller, increased motive force, etc., without any counteracting advantages that can, *a priori*, be detected.

Tangential locking.

826.—We have just seen that, in modern chronometers with a small lifting angle, it is impossible to set the axis of the balance on the tangent drawn through the point of locking. It remains for us to ascertain whether any real advantage is secured by making this locking absolutely tangential.

The gain is manifest if the face of the locking-stone is concentric with the centre of movement of the detent; but, when draw exists, not only is there no such advantage in making this point tangential, but actual inconvenience may result from it.

Three cases may present themselves:

The locking face may be inclined and half of the stone removed, so that the surface would lie accurately on $x\ z$ (fig. 52), the prolongation of a radius of the wheel, or more may be cut away, as $c\ d$ or $i\ g$; the points of contact will in these cases be at b, a and c.

On the first assumption the pressure will act in a direction towards the right of the detent; in the second case it will tend from the locking stone towards the centre of movement of the detent, along the line $a\ h$; and in the last case it will be directed towards the left.

If the resting point is at a, the detent will have no tendency to move to the right or left; and this practically secures a concentric locking face. But there is no advantage in this, because the resistance occasioned by draw must be overcome before the unlocking can be accomplished; a resistance which is mainly due to the height of the small incline that produces the recoil of the wheel (**623**).

The setting of the locking point at b is very objectionable; the resistance opposed to unlocking will be sensibly the same as it was for the plane a, and, since the pressure tends to force the detent from the wheel, the least shake will cause the detent to leave its banking screw. And it is easy to see, by using a

powerful eyeglass, that occasionally a slight interval exists between the two when this condition obtains.

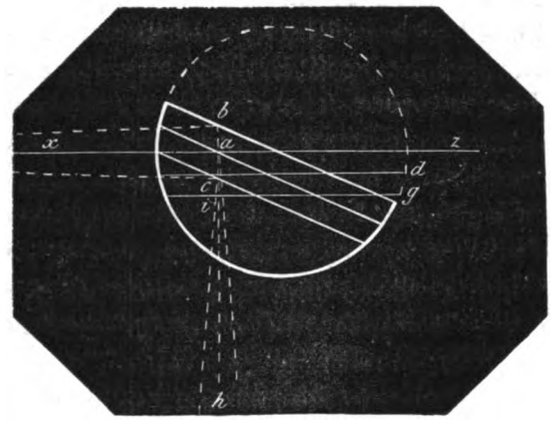

Fig. 52.

The only logical course is to set the locking-point *below* the tangential position or, approximately, at *c*. The steadiness of the detent against its banking being then more secure it will be possible to slightly diminish the height of the incline that causes the wheel to recoil, and, if this is done, rather less of the energy of the balance will be absorbed in the unlocking (**630**).

827.—The height of this small incline gives a measure of the resistance to unlocking.

The direction of the pressure against it indicates the degree of steadiness that the detent will possess during the locking.

By intelligently combining these two elements we shall be able to determine the most favourable position for the locking point.

The Draw.

828.—The resting surfaces in the earliest detents by L. Berthoud were concentric with the centre of movement of the detent, so that the wheel was not made to recoil. At the present day the tooth rests near the base of a straight incline, so that the wheel is impelled slightly backwards at the moment of unlocking.

Some few watchmakers still maintain that draw is useless both in the lever and chronometer escapements; and their main argument is based upon the fact that the pressure takes place in the direction of the centre of movement and that the detent, in virtue of its inertia and state of perfect equilibrium, has no tendency to move either in one direction or the other.

Without instancing the opinion held by a very great number of distinguished makers who have practically recognized the necessity of draw, or the fact that it is at the present day almost universally adopted (although this must have some weight in the discussion of the question), we would point out to the advocates of concentric lockings that, when they deny that movements of rotation in the plane of the pallets or detent have any influence, they are justified, since such motion is exceedingly rare; but they appear to forget or to ignore the fact that the displacement of any form of locking piece by reboundings or quiverings when it is not properly held against its banking, etc., is mainly caused by a shake, or by the reciprocal action that occurs between moving bodies.

Theory clearly establishes this fact, and we have given experimental proof of it in article **635**. An experiment, moreover, that can be easily made will suffice to convince the most incredulous.

829.—If a chronometer or watch with detent escapement be so placed that the balance can be stopped at will while performing the supplementary arc, and if, in several trials, the balance is checked while at the same time the mechanism is subjected to sudden shakes and blows, the observer may perceive, not indeed always but in very many cases, on carefully watching the escapement held up to a cross-light, that the detent is at times held by the tooth of the wheel so as not to touch the banking.

This fault is very common in escapements of Swiss construction, where the draw is secured by slightly rotating the locking stone, cut so as to make the locking tangential; and it is especially noticeable when the detents are at all soft or insufficiently rigid.

This question of draw has been very fully discussed in **622** and the following articles; we therefore refer the reader to them.

An experimental datum, which, however, allows of considerable latitude, fixes 12° as the best inclination for the locking face. The maker must modify this amount according to special circumstances; it will be rather less for marine chronometers and more for pocket timekeepers.

When the draw is somewhat considerable but at the same time the pitch of the wheel and locking stone very shallow, this draw will remain constant for the longest period; (it must, of

course, be understood that the pitch is sufficient to guarantee perfectly safe action).

830.—In practice, when it is required to set the face of the locking stone at the proper inclination, the detent is fixed on a plate made for the purpose and provided with a projecting pin that serves to centre the detent. The ruby is placed in position and, while the shellac is still warm, small tweezers that are centred on the same pin enable the workman to rotate the stone to the amount required, as indicated by a long pointer; this forms a prolongation of the prong of the tweezers that is in contact with the face of the locking stone, and the free extremity traverses a graduated circular arc.

The Lifting angle.

831.—In the detent escapement the lift takes place precisely as in the duplex. The lifting angle must conform to the conditions laid down in **96**, etc., **700**, and **701**; we therefore refer to them to avoid repetition.

In chronometers of modern construction, which are superior to those of older date because a more perfect harmony exists among their several parts, the lifting angle has been considerably reduced. It has a mean value of 45°.

This reduction secures an action that is less oblique to the line of centres; it has therefore made the character of the lead and of the friction more satisfactory, and the escapement is less sensitive to the resistance opposed by thickened oil; especially when the acting surfaces are supplied with oil, or when it has worked up to them.

The Lead.

Form and inclination of the acting surfaces of the Teeth and Impulse pallets.

832.—Arnold's escape-wheel had teeth projecting from the flat, like a crown wheel; and they are rounded so as to act on a straight pallet directed towards the centre of rotation of the wheel, as in an ordinary depth.

L. Berthoud and Motel employed inclined pointed teeth; but the face of the pallet, while being directed towards the centre of the balance-staff, was curved to a certain pre-determined extent where it engaged with the teeth, the radius of curvature being considerable (fig. 10, plate IX.).

Earnshaw's wheel is the same as that of L. Berthoud, except that the faces of the teeth are much more inclined because

the pallet lies in a very oblique plane passing midway between the edge and centre of the roller.

In each case the lifting angle measured about 60°.

Assuming the same diameter of wheels, if we endeavour to ascertain their respective merits we observe that:

Arnold's wheel rests against the locking by a short arm and exerts a proportionately increased pressure. The rounding off of the teeth causes a loss of some degrees in the lead (so that in order to obtain the same lifting angle a larger wheel will be necessary). We thus have an impulse of rather less energy communicated to the balance, and in effecting the unlocking it will be called upon to overcome a somewhat greater resistance.

Berthoud's wheel, with a pallet slightly curved at its extremity, has less drop before engaging and an increased velocity towards the end of the lead, as compared with Earnshaw's wheel, which acts against a straight pallet set at an inclination. In this latter case the tooth falls against the locking stone with rather less energy but it gives rise to a species of draw on the impulse pallet which increases the lateral pressure on the pivots of the balance. The tooth, moreover, can never be so far inclined as only to act with its point; and this accounts for the frequency with which we find the teeth of wheels thus formed to be worn on the face.

833.—The systems of Earnshaw and L. Berthoud are, with reason, regarded as of equal value; they are preferable to that of Arnold because, rather less force being wasted, a heavier balance can be employed, and this diminishes the risk of setting since the unlocking absorbs less energy.

Each of them, although having some faults, possesses important advantages. Modern makers have thus been led to adopt a combination of the two, which has as far as possible the advantages of both systems. The straight impulse pallet is usually preferred, and it is set so as to cut a radius of the roller at right angles, at about a quarter of its length from the centre (figs. 5 & 13, plate IX.).

834.—The inclination of the front faces of the teeth should increase with that of the pallet. As a general rule, in practice the face of the tooth is inclined at an angle of about 27° to the radius of the wheel (A *h z*, fig. 5, plate IX.).

To reduce the weight of the wheel, without at the same time diminishing the acting faces beyond what is necessary

(**41**), it is hollowed out on both sides, or more deeply on one side only. Fig. 53 shows two of the forms (*a*, *n*) of teeth that are employed, the one at *n* being more generally preferred.

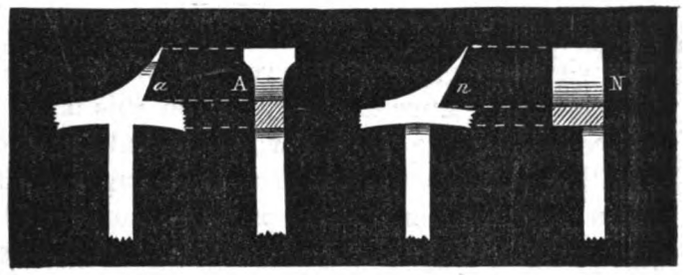

Fig. 53.

A and N are vertical sections of the flat of the wheel, viewing the face of the teeth.

THE DETENT.

On the form of the detent and of the unlocking spring.

835.—Detents and auxiliary springs are to be met with that are bent in the most fanciful manner; their makers flattered themselves that they would secure novel properties in the escapement, whereas they have only succeeded in proving themselves utterly ignorant of the very rudiments of Mechanics. Whatever be the form given to the detent and spring, it is always the virtual length of these levers that has to be considered (**135** & **623**).

This applies to an unlocking spring when it only bends at one point; but if it is thin throughout its entire length or for a certain part of its length, twisted forms, etc., will give rise to flexure that is often very complex and always detrimental.

The auxiliary or unlocking spring.—Its direction.

836.—A steel unlocking spring occasions wear in the course of time; the cause of this has been already indicated in article **810**. It is best to make it of alloyed gold (when it is sometimes known as the *gold spring*) resting against a steel pin or simply against the extremity of the detent.

We believe that the English makers were the first to employ a spring of this metal.

837.—Should the spring be directed towards the centre of the balance-staff? A reply can only be obtained to this question by examining actual escapements.

Three different forms are represented in figs. 10, 13* & 5 of plate IX.; they are shown on a larger scale and detached, in figs. 1, 2 & 3 of plate X., where the details will be more clearly visible.

838.—The lines *e b*, *e d*, fig. 1, join the centre of the balance-staff with the centres of movement of the detent and unlocking spring, and, since the extremity of this latter is at *s*, it will be struck *after* the line *b e* when the unlocking occurs; this action will therefore be unaccompanied by engaging friction. On the return of the balance the spring will only be touched *after* the line *d e* and the friction will still only be disengaging.

Remark.—The great inclination of the spring gives rise to a considerable pressure at the point of contact with the pallet and against the pin, since the spring is, as it were, bent up on itself.

This obliquity of the spring to the body of the detent, besides having the disadvantages mentioned here and in **810**, is objectionable in that it involves an increase in the weight of this detent by the addition of the arm carrying the spring, and indeed by more than this amount if it is desired to distribute the metal equally on either side of the detent.

839.—We will now consider figure 3.

The pallet engages with the spring when on the line *a r* joining the centre of the balance with the centre of flexure of the detent. On returning, the spring is struck on the line *m a*. Both actions will be accompanied by only disengaging friction on condition that the thickness of the extremity *n* of the spring does not exceed the interval between the lines *a m*, *a r* at the point *n*. As this condition involves a point of extreme thinness, there will always occur in practice some engaging friction, according as the spring projects beyond *a m* or *a r*.

When the form of detent represented in fig. 5, plate IX., is employed, the auxiliary spring is set so that it lies slightly to the left, as nearly as possible forming a tangent to the circumference of the balance pivot; and this causes a little engaging friction at the unlocking.

840.—In fig. 2, plate X., the unlocking commences *before* the line *g c* and in the dumb vibration the engagement is also *before* the line *f c*. In both cases the friction is partially engaging.

* By an engraver's error the extremity of the detent is brought too much to the left. It should be nearer to the line joining the point of flexure *m* and the balance centre *a*.

As regards friction then this system is inferior to the two previously considered; if some makers do prefer it, this is doubtless due to the fact that it is characterized by great simplicity, and that it is easy to correct any slight error in the position of the extremity of the spring, by bending or straightening either the end of the detent or of the spring itself.

We must assume that one advantage balances the other, for both systems are adopted by makers who are well-known for the excellence of their productions.

We shall further consider this subject in the following article.

The friction of the unlocking spring.

841.—The fact of the friction being partially engaging when the unlocking takes place does not appear to influence in any way the makers who adopt the forms of detent shown in figs. 5 & 13 of plate IX. (observing the footnote on page 494).

The unlocking should be effected with *absolute certainty;* but, in pitching the escapement, it is almost impossible to guarantee that each face of the spring is accurately on a line of centres. It will be in advance or fall short of it by an amount which, although doubtless very small, is apt to be increased by the movement of pivots in their holes and at times by the quivering of the detent itself.

If the gold spring is struck beyond the line of centres, a given angular displacement of the detent will involve a deeper pitching of the spring and pallet, and thus both the extent of acting surfaces and the amount of engaging friction, when the spring is lifted in the dumb vibration, will be increased.

Many of the best makers allow part of the action to occur before the line of centres, for they consider the advantages and objections of each of the cases under discussion to balance one another, and they further take account of the fact that it is impossible, in practice, to attain to mathematical accuracy, especially in the case of an operation that is so rapid as the unlocking of a detent; the engaging friction thus produced is assumed to be neutralized by advantages in other directions.

The entire question is thus shown to be one of experience; in other words, we must ascertain the nature of the contacts in a large number of chronometers that have been proved by long trial to be satisfactory; but so far as we are aware such observations have never been made on an extensive scale.

All that we can assert is that several chronometer makers deny that they have ever been able to detect traces of the influence of the unlocking pallet more frequently on one system than on another.

The detent must bank at its centre of percussion.

842.—Some makers appear to attach but little importance to the exact determination of the point at which a detent is banked. We think they are wrong in doing so; for if, when the point is improperly placed, they also have a detent a trifle thin, liable to quivering and unequally elastic at different parts, etc. (and this is by no means of rare occurrence), anomalies may exist whose cause will be long sought in vain.

If the banking screw is carefully located the detent will expend, in striking it, all the force with which it is impelled and will at once become stationary. On the other hand, if the banking does not occupy its theoretical position, the detent will quiver and will engage with the teeth when pressing against the banking with a variable force.

843.—The point or centre of percussion (for a given distance between the locking stone and balance) varies according as the detent is long or short; it follows that those makers who set the banking screw so that the detent rests with the locking stone against it, cannot make the detent of an arbitrary length: both this length and the general distribution of the material constituting the moving portion of the detent must be such that the point of percussion coincides with the banking point.

Very many escapement makers, whose only aim is to make a servile imitation of the works of their superiors, demonstrate in this one point their complete ignorance of a simple mechanical principle.

By using a very delicate micrometer, all the dimensions of a detent may be taken, a drawing can be made and the position of the banking determined by calculation.

We thus obtain a good first approximation. It may be experimentally verified as follows.

Employing an eyeglass of high magnifying power and conveniently situated, it is easy to ascertain whether the detent quivers on striking against the banking screw; but this verification must be made with the plate held in several different positions, first horizontal and then vertical.

If the whole is well arranged, no difference will be detected in the movements of the detent however the position be altered.

The Banking Screw.

844.—The detent must not be held against an absolutely rigid body (**35**).

The two methods of banking shown at *c*, fig. 7 & *d*, fig. 8 of plate IX., as well as those in which the detent rests against the head of a screw or a cam screwed to the plate of the movement, systems that are much used in Switzerland, are unsatisfactory.

The cam presents a too great surface of contact, and the slightest angular motion changes the position of the point of banking.

A screw head carrying a pin as arranged by L. Berthoud (*a*, *a'*, fig. 10, plate IX.) is good; but only if we avoid rotating the screw (for the purpose of adjusting the pitch of the wheel and locking stone) as then the position of the banking point is altered.

It is better to fix a banking screw in an arm projecting from the foot of the detent. This arrangement, shown at *g*, fig. 13, & *h*, figs. 5 & 6, plate IX., is the same as that adopted by Breguet; but less massive and the cam is removed.

A pin set in the head of a screw as well as an arm carrying a banking screw must be of but moderate volume; if completely wanting in elasticity they would in time wear at the point of contact; on the other hand, if made too thin they will oscillate at each blow of the detent. Experience must decide as to the most convenient dimensions.

Diameter of the locking stone.—Angular movement of the detent.

845.—When the locking takes place at the middle of a ruby cemented into the detent, the radius of the cylindrical face of this ruby is determined by the angular movement of the detent in effecting the unlocking.

At the present day the locking does not occur at the middle of the stone and it is necessary to slightly increase the diameter. This diameter is generally made about one quarter of the interval between the points of two teeth. A large ruby needlessly loads the detent, and, if its corner is not considerably bevelled, it renders a greater angular motion of the detent essential; in other words, it would be liable, on returning to its position of rest, to strike against a tooth with its curved surface.

846.—In a well-made marine chronometer escapement, that is not characterized by heaviness in any of its parts or resistances due to bad arrangement as a whole, the angular deflection of a spring detent measures about 2° (brought about by an angular movement of about 22° at the balance, inclusive of the action on the gold spring during the dumb vibration). This amount of lead may be divided into two portions differing somewhat in extent; the first effects the unlocking and the second draws the detent aside to such a distance from the wheel as to ensure that the tooth just released has sufficient time to advance out of reach of the detent returning to the banking.

It is hardly necessary to observe that, for a given balance movement, the angular path of the detent increases as the centres of motion are brought nearer together.

A light detent or one that is at all stiff requires that the angular movement *after* the unlocking should be of rather greater extent than is necessary with a heavy or weak detent of the same length.

It is impossible to give definite figures in relation to the subject for detents are found to vary in this respect. Recourse must be had to experience and observation as the best guide (**854**).

Long and Short Detents.

Experimental Data.

847.—Let *c n* (fig. 5, plate X.) be a detent with its centre of motion at *c*, the locking of the wheel at *b* and the extremity of the gold spring at *n*. Assume the unlocking pallet *r* to be centred at *a*.

The angular displacement of the detent will be *n c g*; and the centre *b* of the locking stone will be displaced to *x*.

Now consider the centre of movement to be transferred from *c* to *d*. The angular motion will become *n d s*; that is to say, the displacement of *n* will be somewhat greater, and the centre of the locking stone *b* will be carried to *y*.

The friction occurs through a longer period but the angle is rather less oblique, so that we may regard the friction in the two cases as the same. It will be noticed, however, that the greater displacement of the locking with this detent, which is necessarily heavier than the other, requires that it return more rapidly to its banking and therefore that its spring be more stiff. To avoid this increase in the resistance to unlocking, the pitch of the detent and pallet must be diminished so as not

to exceed the arc ij, if it is required not to displace the locking further than x; a distance that is assumed sufficient in the case of the detent $c\,n$.

We thus avoid one fault only by falling into another, for, if the pitch of the detent, $c\,n$, with the pallet were no more than is requisite to secure certainty in the action, then the pitching of the longer detent, $d\,n$, would be insufficient. It would, moreover, be liable to much greater variations through changes in relative positions due to play of the pivots.

848.—From the theory of the lever we obtain the following results:

Detent $c\,n$.—Power arm $c\,n = 4$
Resistance arm $c\,b = 2$
Detent $d\,n$.—Power arm $d\,n = 8$
Resistance arm $d\,b = 6$,

which proves that the long detent offers the greater resistance to unlocking.

On the whole, then, the long detent has two advantages: a less rapid progressive increase in the elastic resistance or tension and a less oblique action against the pallet; but, at the same time, it is objectionable in offering a greater resistance to unlocking, and in being heavier, less firm, and less certain in its action; so that the faults far outweigh the advantages.

This demonstration shows us what an error is committed by the Swiss makers, who form the detent of immoderate length under the false impression that they thereby diminish the resistance to unlocking.

849.—Consider now the case of a short detent and assume its centre of movement to be at t; the length will be $t\,n$.

Unlocking under the assumed conditions will be impossible, and to bring the point b to x, the detent must be prolonged to k.

A glance at the figure will show that this detent, $t\,k$ or $t\,z$, will involve a much greater effort on the part of the balance than does the detent $c\,n$, owing to the great loss of force occasioned by the deep pitching, necessarily rapid bending of the spring, etc. We have already said enough on the subject; it seems useless to insist on this point.

It is, then, evident that those discussions as to whether a long or short detent is preferable are as utterly idle as in the case of the long and short balance-springs, levers, etc. The horological art does not admit the existence of long and short

detents. It fixes for each particular escapement the length that secures an absolute certainty in the several actions, and which offers such resistance to the balance as shall have the least possible disturbing effect on its performance, and therefore on its regulating power.

850.—In the case of the marine chronometer as now made, it has been experimentally ascertained that the acting length of the spring detent should not be less than the diameter of the escape-wheel, nor more than the radius of the balance.

A pivoted detent, measured from the centre of motion to the extremity of the auxiliary spring, should be much shorter (by rather more than one-third this amount); for otherwise its weight, being increased by that of a counterpoise, would render the action of the escapement uncertain, especially in pocket chronometers beating 18,000 or 21,600 vibrations per hour.

On advancing the locking stone by one tooth.

851.—As a rule the escape-wheel rests against the locking stone by the second tooth in front of the one which will give the immediately succeeding impulse to the balance.

Would there be any advantage in making it lock against the third tooth?

The preceding article has already settled this question.

The choice of the third tooth renders a long detent essential. For, if of moderate length, the portion $n\,b$ (fig. 5, plate X.) becomes considerable in comparison with $b\,c$, and such an arrangement entails all the inconveniences that are involved in an excessive angular movement, etc.

Locking by the third tooth can have no advantages except in the case of a wheel of higher number or different diameter than those in ordinary use. At the present day such a system is only of service under certain special circumstances, and a watchmaker who is thoroughly conversant with the theoretical principles and possesses sufficient practical knowledge, will be able to decide for himself without difficulty.

Impulse Roller.—Unlocking Roller.

852.—The size of the impulse roller depends directly on the number of teeth of the wheel and the lifting angle (**824**).

For a given wheel it varies directly with the lifting angle.

Its diameter as compared with the other dimensions of the escapement may be determined by means of a large scale draw-

ing or, as is more usually done, by tracing the calliper of the escapement (**866**).

The diameter of the unlocking roller is usually from a quarter to a third of that of the impulse roller.

Number of Vibrations; determining the amount of angular motion of the detent.

853.—Marine chronometers with detent escapements have been made to beat 21,600, 18,000 and 14,400 per hour.

The last number has been generally preferred, and this circumstance can be explained by the fact that with a higher number, 18,000 for example, and all the other parts to correspond, it is necessary to employ a lighter balance, impart a more rapid motion to the wheel, increase the lifting angle as compared with the supplementary angle, and, lastly, to multiply the number of wheels in the train or increase their angular velocities.

Makers have, then, done wisely in not exceeding 14,400 vibrations in the case of marine chronometers, where the parts are somewhat large and heavy and therefore render a powerful balance essential. In the case of pocket chronometers the comparative lightness of the mobiles makes it advantageous to use the number 18,000 or 21,600, as by so doing the effects of shakes, etc., are rendered less detrimental.

854.—The adoption of higher numbers, moreover, enables us to diminish the extent of the angular movement of the detent, for it must return more promptly to the banking screw.

In marine chronometers this motion, varying from 2° to 4°, is effected by a balance movement measuring rather less than *half* the lifting angle (**846**).

In pocket timekeepers with this escapement, that give 18,000 vibrations, it need not be more than about one-third, and, for 21,600, one-fourth. These figures have been obtained by some of the best makers from extended series of observations; but it must be remembered that they cannot be regarded as in any sense absolute.

Is the spring detent superior or inferior to the pivoted detent?

855.—This question has been very often asked and has given rise to long controversies between some of our most skilful chronometer makers; several interesting articles have appeared in Vol. III. of the *Revue Chronométrique* on the subject.

Without being in any way partial or desiring to do more

than determine which is preferable, we shall discuss it, holding the opinions of our eminent fellow-workers in all respect, but at the same time being independent, as we can be from our complete disinterestedness.

These observations will necessarily be brief; for very many points in the discussion have already been settled by the preceding explanations.

Here in a few words is the history of the subject.

The pivoted detent, invented in France by P. Le Roy and perfected by L. Berthoud, was experimented upon in England, mainly by Arnold in his chronometers without the fusee. Fig. 8, plate X., represents one of Arnold's detents, taken from a drawing by Tavan, who copied it from an English chronometer.

This form of detent is radically wrong. The length from *d* to *o* is enormous as compared with that from *o* to *e*: it requires a considerable angular movement to effect the unlocking, for otherwise the pitch of the wheel and locking stone would be insufficient to ensure certainty of action.

The banking screw *s* is too near the centre of motion; the quivering occasioned by the weight and the great length of the arm *o d* will cause the detent to engage with the tooth of the escape-wheel when resting sometimes firmly, sometimes lightly and at times even not at all against the banking screw. This effect will be the more serious and difficult of detection according as the point of banking is brought nearer to the centre of movement, and as the pivot-holes are larger and allow of a lateral displacement that will vary according to the state of the oil.

If to these remarks we add that the English workmanship of that day was much less delicate and more massive than that of French chronometers, it will be evident why detents of this construction were irregular in their action and became sluggish in the course of time. Dissatisfied with the results arrived at, Arnold assigned to the resistance of oil effects that were mainly due to faults of construction and to the general mechanical arrangement. Compare Arnold's pivoted detent with that of L. Berthoud (*e' o' d'*).

We would observe that Arnold's recovering spring was attached to a collet set concentric with the axis.

His arrangement of the escapement with spring detent (fig. 7, plate IX.) is infinitely superior; this accounts for his having abandoned the experiments on the pivoted detent.

The pivoted detent is undoubtedly a French invention; but this is no proof that we must give the entire credit of the invention of the spring detent to England. We have set this historical point in its true light in article **822**.

Advantages and disadvantages of each form of detent.

856.—It is urged against the spring detent that:

It offers a greater opposition to the unlocking;

This resistance varies with the position in which the escapement is held, for it is supplemented by the weight of the moving part or diminished by this amount according as the escapement is held vertical or horizontal;

It is apt to be strained.

857.—The objections to the pivoted detent are:

Its necessarily greater weight;

The increased friction on the gold spring, since the pitch is deeper and the angular movement greater;

The resistance at the pivots;

Lastly, very great care is necessary in its adjustment and in determining the position of the banking screw. The most careful workman is often obliged to pitch the detent two or three times before arriving at the exact position.

858.—The spring detent is advantageous in that:

It permits of the force being more advantageously distributed;

The difference in the resistance opposed in various positions cannot come into play in the case of marine chronometers;

With a broad flat spring, such as is employed at the present day by the best makers, distortion, even if it does occur, has no sensible effect.

859.—The defenders of the pivoted detent urge that:

It is more solid and there is no possibility of this straining;

There is no change in resistance when the position is varied;

It is more efficient in diminishing the detrimental effect of the fall on to the banking screw (**861**).

As to its being more easy of construction we cannot grant this to be true. Both systems of detent require the hand of a first-rate workman in their construction. Such pivoted detents as *can* be made with ease are bad and heavy; they are simply useless.

860.—Ignoring the question of the resistances that depend on the form of recovering spring and the presence of oil on the pivots, points which will be discussed in the two succeeding

articles, we think very little of the fact that an unequal degree of force is required to effect the unlocking, just as we regard the difference as to difficulty of construction of no moment; for this inequality, which is very slight and mainly results from the difference in the lever arms, is partially neutralized by the somewhat greater friction owing to the lead being of longer duration; and the want of equality thus often quite disappears.

The influences that have to be considered are almost instantaneous, and the practical differences, in the case of good workmanship, slight, and we have come to the conclusion, after consulting experienced makers in order to ascertain which offered the greatest facilities of manufacture, that the spring detent is justly preferred to the pivoted detent for marine chronometers (with 14,400 vibrations); and that the pivoted detent offers more advantages (**853**) in the construction of pocket chronometers (with 18,000 vibrations).

In conclusion, they are two distinct mechanisms having valuable qualities of different kind, but they likewise have objections that differ. In the hands of a skilful chronometer maker equally good results may be secured, and success may be relied upon with either form. He will know how to make the most of their good points and diminish their faults; we must therefore not make either system responsible for errors committed by unskilful imitators.

The friction at detent pivots.

861.—A pivoted detent does not rotate on its axis but merely oscillates through a very short arc. The contact between the pivot and pivot-hole gives rise to a simple pressure or a very limited rolling. This action, then, cannot be supposed to resemble the true friction at the other pivots of the mechanism.

The side of the hole is subjected to an elastic pressure analogous to that exerted on the head of the banking screw; as in that case, too, the elasticity of the support (the staff of the detent) diminishes the effect of the fall. Two very remarkable consequences follow from this fact: (1) the face of the *steel* banking does not deteriorate, although there is a constant succession of impacts on the same point, which cannot be said of the spring detent; (2) the pivots of the detent and the sides of the pivot-holes maintain their initial condition.

Seeing then that, when the locking stone is of ruby instead of steel, the points of the escape-wheel teeth suffer no wear, and

that the pivots of a well-made pivoted detent remain intact, two results follow: firstly, it is unnecessary to apply oil to the locking stone, and, secondly, the oil applied to the pivots will remain good. This latter fact is rendered the more certain by the circumstance that these pivots are only subjected to a slight pressure, without sensible friction; and the oil is not continually being intermixed by rotation, as in the case of other pivots.

This preservation of the oil is now a proved fact; it is therefore not usual to employ jewelled pivot-holes.

862.—The resistance opposed by the detent pivots to motion has its source, not in friction properly so called, but in the effects of adhesion and capillarity. We are probably not far from the truth if we look upon it as analogous to the friction that opposes the commencement of motion of a stationary body.

What is its value? It is impossible to decide this question *a priori*, and the experiments it would involve are of so delicate a nature that we cannot reasonably expect an exact figure. It appears certain however that, whatever its value, it varies in time, for the resistance opposed by adhesion to the separation of two bodies is in proportion to the extent of surfaces in contact and the consistency of the lubricating body between them.

If the detent pivots are fine and not too long, this resistance has been considered to be inappreciable by some eminent makers, notably by H. Robert and the younger Berthoud.

At the same time, small though it may be, it certainly does exist; and, without making either too much or too little of it, since a suitable recovering spring can easily counteract its influence, it should not be entirely ignored (end of **863**).

Straight and spiral recovering spring.

863.—Moinet, in comparing the two forms of detent, observes that the pivoted detent would appear to offer a less rapidly increasing resistance, since the action of the spring detent depends on a short straight spring, whereas in the other a relatively long spiral is employed.

We will assume for the present that such is the case.

Suppose that to the same detent recovering springs have been applied of the two kinds, and of equal force, that is to say such that, when deflected through the same angle, they maintain the detent against its banking screw with equal steadiness.

For a given angular displacement of the detent, the resistance will become greater as the length of the spring is diminished.

The velocity of its return will, then, vary approximately in inverse proportion to this length.

Would it then be better that there should be less resistance during the lead, involving a diminished velocity in the return? Or is it preferable to have a somewhat greater resistance opposed to the unlocking and, at the same time, a more rapid recovery.

A series of comparisons, continued for a sufficiently long time, in which different recovering springs were employed in conjunction with the same detent, might decide these questions. It would be imprudent to attempt their solution solely from a theoretical point of view and with our present knowledge; the results might easily be erroneous, for there are frictions, flexures, etc., that are so rapid and minute as to escape detection, which cannot be represented by even approximate figures.

We have assumed that the resistance increases more rapidly with the straight spring and this point could be easily settled with angles of deflection of some extent, but, the angle in this case being *very small*, 2° or 4°, the straight spring may be compared to a spiral spring of sufficient strength and suitably arranged.

If it were demonstrated that the velocity with which a detent is brought back by a spiral spring diminishes with time, it would prove that the resistance opposed by the pivots ought to be taken into account.

864.—Although these questions still remain in abeyance, the following conclusions are accepted as true at the present day.

Observation has proved that a straight spring is sometimes oxidized at the point of contact and still more frequently adhesion occurs that has a sensible effect on the very limited movement of the spring; in many cases both these circumstances will in time check the return of the detent.

A carefully fitted spiral spring gives rise to neither friction nor lateral pressure, and its action may be relied on if it possess the strength necessary to restore the detent to its position of rest with sufficient rapidity; in this case the resistance opposed to unlocking is approximately the same as with a spring detent, assuming all other conditions to be the same. In using a spiral recovering spring, then, we cannot count upon the slight difference in the action due to the less resistance offered to unlocking by a pivoted detent; an advantage, by the way, which we have always regarded as insignificant.

The spiral spring should not be longer than is necessary to

secure a motion fairly concentric with the pivots (3 to 5 turns). As the length is increased it will become more sensitive to variations of temperature and to shakes.

To design the Escapement.—To calculate its proportions.

865.—The methods and details given in **749** and the following articles are applicable here, and we have given such exhaustive explanations that it seems needless to do more than give the directions for tracing the calliper; these will be found below.

The same may be said as to calculating the proportions of the escapement when the wheel is known. The details of articles **748, 754,** etc., will suffice.

CHAPTER III.

PRACTICAL DETAILS.—CAUSES OF STOPPAGE AND VARIATION.

To draw the calliper of the escapement.

866.—Through a perfectly smooth brass plate drill a fine hole (*a*, fig. 7, plate X.) for the centre of the escape-wheel, and trace out the circumference. Assume the wheel to have 15 teeth.

This plate is centred on the wheel-cutting engine by the hole *a* and waxed on the table; then draw the lines *a c i* and *a b*, with the pointed cutter, inclined at an angle of 24°, and draw *a d* at 24° to the second line.

Lastly draw *a m*, which accurately divides this second angle into two equal parts. The centre of the balance will lie on this line.

The lift, inclusive of the drop, having been previously fixed upon, say 50°, two methods may be adopted for ascertaining the position of this centre.

First Method.—Draw with very great care the angle *q z p* (fig. 6) of 50° on a piece of metal, and accurately bisect this by the line *x z*. The portion must then be cut out, of the form shown in fig. 6, and placed on the calliper so that the line *x z* coincides with *h a*. Maintaining this coincidence, let the sector *q z p* slide from *a* towards *m* until its two sides pass through the points at which *a d* and *a b* cut the circumference: the angular point then fixes the point *m*.

A spot is made there strictly coinciding with the apex of the sector and fixing the centre of the balance.

Second Method.—If a line be drawn through *d* and *b* (fig. 7) it will be observed that the triangle *m d b* is isosceles; so that, the angle *b m d* being 50°, each of the two other angles of the triangle will be 65°. Hence, by drawing through *b* and *d* lines inclined at an angle of 65° to *d b*, the point of intersection of the lines *b m*, *d m* will give the required centre.

Or this centre might be determined by calculation; but those who are in a position to employ this method will not require further details than those above given.

The angle is verified by a protractor fitted with an index and centre pin, or on a wheel-cutting engine. If any error is detected, it is corrected by moving the centre towards or from the point *a*, and a fine hole is then drilled at *m* on the drilling tool.

From this centre draw the circumference of the roller. It should be at such a distance from the points *d* and *b* as to allow of the necessary play and secure the proper action of the wheel and roller.

Draw the circumference of the unlocking roller; its diameter is between a quarter and a half of that of the impulse roller, varying according to the size the balance-staff is required to be.

867.—The most delicate operation in the entire tracing of the calliper is the determination of the centre of the locking stone.

We know that this locking should occur a little in advance of the tangential position (**826**). In order to satisfy this condition draw *a s* inclined to *a c i* at an angle of a few degrees; through this line *a s* draw, from the point of contact of the tooth *c*, a perpendicular *c r*, which fixes the direction of the body of the detent.

Draw through the point of contact *c* another line *c t* forming an angle of about 12° or 15° with *a s*, and the centre of the cylindrical face of the locking stone will lie on *c t*; it must be remembered that the pitching (generally amounting to from one-quarter to one-third the diameter of the stone) should not be more than is absolutely necessary to render the several actions certain (**829**).

Drill a hole to indicate the position of the locking stone,

increasing its diameter as required; it is then easy, by examining the finely drawn lines with a powerful eyeglass, to make sure that the hole is properly located.

Drill the hole for the detent foot screw, and draw a line to indicate the direction of the unlocking spring, etc.

When the calliper has been completed in the manner explained, a well-proportioned escapement must be made, using it as a guide; the maker ascertains that the parts are properly proportioned by causing them to work together when placed in position on the calliper.

868.—*Remark.*—The wheel-cutting engine is the most accurate instrument for the measurement of angles, but in its absence the circumference of the wheel may be traced on the plate of the calliper and then, placing the wheel itself on the plate, mark with very great care the points of three successive teeth by fine dots. The calliper must then be completed by means of a well-made ruler and compass. The accuracy of the drawing will be increased by making it on a plate of considerable surface so that all the lines can be prolonged; the chances of error will of course diminish as the divergence of lines becomes greater.

PRACTICAL DETAILS.

869.—We shall be brief under this heading, confining ourselves to such points as are peculiar to the detent escapement. For any watchmaker that undertakes to make it must be assumed to possess sufficient practical knowledge, and to have already had experience in other forms of escapement. The manipulative details are in great part identical with those already given so fully either in the preceding chapter or in our discussions of the duplex and lever escapements, to which the reader is referred.

The Escape-Wheel.

870.—The wheel is made of well-hammered brass or of an alloy of gold, silver, and copper. It is hardened by a suitable annealing. It is cut with a circular cutter or, more frequently, a single rotating or hooked cutter as in the case of duplex or lever escape-wheels. Fig. 16, plate X., shows, in section, the forms of the two cutters; one of these is used to cut out the spaces and to form the front faces of the teeth, and the other to make the back of the required shape.

The tooth must not terminate in a sharp angle but in a small rounded surface. (See the articles on the duplex and lever escape-wheels.)

Most makers are content to gently rub the points of the teeth with a piece of oiled wood, after the escape-wheel has been set in position in the chronometer.

To make a spring detent.

871.—The most difficult case that can present itself is the replacing of a broken detent. For then its position is rigorously determined.

On a finely smoothed plate mark all the centres of movement with very great care; also the holes for steady-pins, screws, etc.: then draw the several lines (if possible on the wheel-cutting engine) including the direction of the detent. Enlarge with care the escape-wheel hole until its pinion enters the hole, where it must turn truly and with gentle friction.

If the pinion is at a distance from the wheel, turn (on the mandril) a hollow in the plate, which the wheel just enters, being held by the slight contact of the teeth.

Having now set the wheel in its position on this calliper, adjust the roller on it with a staff or arbor passing without play through the enlarged hole corresponding to the balance. It then becomes easy to verify and, if needful, to correct the position of the locking stone (**867**).

872.—Very great care and prudence are essential in the selecting and working of the steel of which the detent is made, for otherwise there is likelihood of the expenditure of much trouble to no purpose. Square fibrous English steel, exhibiting a fine grain when fractured, close-grained and of a grey silvery shade, is most usually preferred. It must be annealed and then hammered, the blows being slight and always directed on to the face of the plate. It is then tempered to a blue shade and, after filing it up square and perfectly flat on each face, mark the several holes for the locking stone, etc., and drill them on the drilling tool; then shape the detent with a file.

The filing of the spring requires close attention lest the metal be in any way strained. The position of the detent should be varied during the operation so that the metal may be removed evenly, for the hand always has a slight tendency to lean to one side.

The spring must be no stronger than is necessary. It should be smooth and gradually increase in thickness from the foot.

The lantern spring, having an opening in the middle, is bad. The elasticity of the two sides is rarely the same; they are nearly always unequally hardened, and their centres of flexure are therefore seldom on the same axis.

873.—The hardening of a detent requires not only the most minute precautions on the part of the workman, but it is also essential that he possess considerable skill in conducting the operation, to avoid distorting or over-heating the spring. The hardening may be accomplished before the requisite thinness is attained; finishing it subsequently with an iron and oilstone dust.

Every escapement maker has his own mode of hardening, the success of which is solely dependent on skill. One of the processes enumerated in paragraph **496** may be adopted, or the following modification.

Make an oblong box in platinum foil, about 5 or 6 millimetres (0·2 ins.) in diameter and longer than the detent, closed at one end. It must be provided with a base to fix it on the charcoal and this latter should be cut away so that the box is supported by its two ends. After introducing the detent, it is raised to a red heat by directing the flame against the platinum jacket and, when the whole is heated to the required temperature, the detent is allowed to slide into oil.

Some workmen previously direct the blowpipe flame on to the surface of the oil at two or three points, and they assert that such a practice prevents any distortion of the detent. We have never tried this method and are therefore unable to speak as to its efficacy.

The detent must not be enveloped in powdered charcoal or animal black: for particles would adhere to the spring at certain points and cause those portions of the metal to receive a less degree of hardness.

874.—When the detent is finished and the auxiliary spring attached, fix a temporary locking stone of steel in position. If the whole has been carefully made it will be possible to correct any error of position by slightly varying the inclination of the locking face or its pitching with the wheel, and the adjustment should then be complete. The steel locking piece will, after all corrections have been made, be given as a model to the lapidary.

Drill holes for the steady-pins of the detent foot, after

placing it so as to be maintained with sufficient force against the banking screw.

Pivots and Pivot-holes.—Balance-Spring and Balance.

875.—The details necessary on these points have been already given under the same headings in Chapter III. of the Lever Escapement. The reader must refer back to those articles and he should be competent to select for himself what is applicable to the present case.

876.—As the isochronal spring and compensation balance are absolutely necessary with this form of escapement, we refer to the chapters in the Third Part of the work that specially treat on these subjects and on Timing.

We would, however, at once mention that the position adopted by many chronometer makers for pinning in the spring, because it diminishes the risk of setting, is that which places the discharging pallet in the position shown in fig. 5, plate IX., for zero tension of the spring; so that a very slight motion of the balance will effect the unlocking.

This rule, although sufficient, is vague; and the following is more precise.

The spring should maintain the balance, when at rest, in such a position that on turning it either to the right or left, it has to describe the same arcs to permit the detent or the unlocking spring to escape.

Causes of Stoppage and Variation.

877.—The detent escapement does not admit of any second-rate workmanship. All the causes of stoppage and variation result from bad manipulation or errors in principle.

It is useless to enumerate these causes, which we have indeed already indicated. Any watchmaker of ordinary intelligence should be able to detect them; otherwise he must proceed to study the escapement from first principles before he undertakes either to make or repair it.

With regard to the faults that are inherent in this particular mechanism, such as *setting*, *tripping*, and *banking*, they have been reduced to a minimum, and need not be feared in a well made timekeeper, except when it receives a jerk during winding or if it is subjected to violent shaking, as in walking, riding, etc.; but we must again repeat that the chronometer

escapement is not suitable for daily use; it should only be employed in instruments intended for scientific observations and by those that know how to take care of it.

Nevertheless we would add, in conclusion, that the chronometer maker should attach very great importance to: determining the effects of rebounding and shaking, especially sensible in the case of light detents;—the condition of the spring of the detent which may give rise to complex and unlooked for effects of flexure, according as it is directed more or less approximately to correspond with the pressure on the locking stone, as it is more or less straight, or curved crosswise in the polishing, etc.;—banking screw out of place;—the velocity with which the detent returns to this screw variable with temperature; etc., etc. In a word, all the several actions must take place *with absolute certainty*, notwithstanding any changes that may be brought about by the age of the oil and alterations of temperature.

878.—We will conclude by referring to the necessity of a perfect equipoise with the pivoted detent, made sufficiently evident by the experiments described in article **635**; we would also draw attention to the rebounding action, to which detents banked against their poise end are more especially subject when the banking screw is not exactly in its proper place. These influences have been more especially marked in pocket chronometers without fusees, and they are attributed, rightly or wrongly, to the excessive force applied to lift the detent. This conclusion has been due to the fact that, on examining the escapement against the light with a powerful eyeglass, in several cases the interval between the detent and its banking when a rebound occurs was found to be most marked with the mainspring fully wound up.

Why was Arnold's detent abandoned?

879.—In the arrangement adopted by Arnold, the pressure of the wheel against the locking stone tended to elongate the detent; in Earnshaw's, the tendency was to press it towards the foot. The latter system has been retained.

It is, nevertheless, inconvenient in that, other things being equal, it renders the steadiness of the detent less certain: the spring of Earnshaw's detent is therefore usually made somewhat thicker than in Arnold's.

But, if the plan adopted by the latter authority is preferable

from this point of view, it has the grave objection of being only available with the teeth projecting from the flat of the wheel, as in those made by Arnold himself. For a given diameter such a wheel would be heavier at the circumference than a wheel with pointed teeth, and would give a less lifting angle; it would exert a greater pressure on the locking stone and this would be as objectionable as the stiffness of Earnshaw's spring; and, lastly, the wheel must be made true with the greatest possible care.

DOUBLE-WHEEL CHRONOMETER ESCAPEMENT.

880.—This escapement, shown in fig. 12, plate X., was first made by Owen Robinson, an English chronometer maker. Shortly afterwards U. Jurgensen* made it with success, and he brought it to a high state of perfection.

At first two superposed wheels were used. They underwent a similar modification to those of the duplex, and are now combined into one; triangular prismatic teeth, projecting from the flat of the great wheel, replace the small wheel.

The use of two wheels is advantageous in that it facilitates the setting of the impulse wheel in position, but their total weight was greater than that of the single wheel, which, moreover, only requires as much care as is devoted to the cutting of the best duplex wheels.

881.—As will be evident from the drawing, this escapement only differs from those already described in having a double wheel. The impulse is applied by an arm D *s* that is shorter as compared with the resting arm. The pressure on the locking stone should be sufficient to ensure the steadiness of the detent, and, if we assume this pressure to equal that of the escape-wheel in an ordinary chronometer, it necessarily follows that the lifting action in the double-wheel escapement will be more energetic than in that of ordinary construction; but it should be observed that this slightly increased pressure, which may be considered beneficial, is opposed by the inertia of a heavier wheel, absorbing a greater amount of energy.

With a 15-toothed wheel, the balance will be planted

* Urbain Jurgensen, a Danish chronometer maker, was born in 1776 and died in 1830: he associated himself with the best makers of his day, and especially Breguet. He is justly celebrated for the construction of excellent chronometers and very good astronomical clocks, and has left us a work on the exact measure of time that can even now be studied with advantage.

somewhat nearer to the tangent passing through the point at which the tooth rests.

On the whole then, balancing the advantages and objections, they very approximately neutralize each other, and the final result or, in other words, the uniformity in the rate attainable, is simply equal to that of a well made ordinary detent escapement.

Jurgensen frequently employed this escapement (with a 12-toothed wheel) in marine chronometers and obtained good results; but, if equal results are obtained, we prefer the ordinary detent escapement: it is more simple and presents greater facilities of manufacture.

DETENT ESCAPEMENT WITH RECOVERING PALLETS.

882.—The detent in the first detached escapement of P. Le Roy was not restored to its position of rest by a spring; its movement was effected by a kind of unlocking pallet attached to the balance and engaging with two small arms carried on the detent staff.

Since his time another form of pivoted detent has been adopted, the detent being brought back to its resting position by the wheel engaging with a supplementary pallet known as the recovering pallet.

Fig. 11, plate X., represents such an arrangement.

The detent *p a* carries an auxiliary spring *f a*. When the roller turns in the direction indicated by the arrow, the unlocking pallet strikes the detent *a*, releasing the tooth *d*. The wheel in its motion strikes the pallet *b* with the tooth *c*, and thus communicates the impulse to the balance. Towards the conclusion of this impulse, the tooth *f*, meeting with the recovering pallet *s*, forces it backwards, restoring the detent to the position it occupies in the figure; this tooth, *f*, is then held stationary in the position *g*.

The return vibration is *dumb*.

883.—Escapements of this class have always been unsatisfactory and for evident reasons.

The recovery is effected suddenly through the action of a wheel, which is moving with a variable velocity; there must, then, necessarily occur a rebound, vibratory movement, etc. These effects are highly detrimental, all the more so because

the detent rests for an instant, so to speak, in mid air; for it only actually rests against its banking *p* during that portion of the dumb vibration which commences with the lifting of the auxiliary spring by the unlocking pallet.

Sufficient accuracy has been obtained for ordinary purposes with some forms of escapement with recovering pallet, but in chronometers used for scientific observations they have given results inferior to those obtainable with a good detent escapement.

884.—If the recovering pallet is set too near the centre of motion, it renders necessary an increased motive force, for it absorbs a great portion of the energy of the wheel; if placed too far off, it will render the detent more sensitive to the influence of the variable velocity of this wheel.

NOTES

ON CERTAIN DETACHED ESCAPEMENTS.

Pin Escapement (for watches).

885.—The pin escapement was invented by Amant prior to 1740. It is believed that Robin was the first to arrange it so as to be detached from the moderator throughout a great part of the oscillation, and to adopt it in watches; but we have no precise knowledge as to this point.

The escapement was made at Geneva at the beginning of the century, and the model made by Tavan, and described in the memoir enumerating his works, dates from that period; a few watches of this construction are still occasionally to be met with.

It was very soon abandoned; for the pins do not retain the oil and an escapement that is not worn is very rare.

It is merely a lever escapement in which pins take the place of the teeth of the wheel, thus rendering it possible to bring the two pallet-arms nearer together. We shall dispense with a description as an examination of fig. 9, plate X., will make its form and mode of action evident.

The pin escapement of watches when compared with the lever escapement as now made is as much inferior as the hook escapement is inferior to the cylinder, and for the very same reasons. The observations contained in paragraphs **585** and **586** are therefore equally applicable to it.

If the pallets are fitted with rubies, and more especially if the pins are formed like those of Savoye's virgule escape-wheels, it possesses a good rate, but in no way better than that of a well-made lever escapement. With equally satisfactory rates this latter has the advantage of retaining the oil better on the acting surfaces, and it can be made by machinery at the present day with very great precision and at low cost (**757**). It is as well, then, that the pin escapement was abandoned.

886.—During the past thirty years several watchmakers have taken out patents reviving this mechanism, which was new to no one but themselves; and one of them has even gone so far as to suggest the use of hardened steel pins, held firmly in the rim of the wheel. To employ a watch wheel so made at the present day is to utterly ignore all the resources and appliances of modern industry.

The arrangement shown at fig. 9, plate X., is the design that has most frequently been followed. In the large model made by Tavan, the wheel had only 12 teeth. To avoid excessive thickness of pallets and to make the whole lighter, he cut away the extremities of the two arms in the manner indicated by dotted lines; but this removal of useless metal did not in any way alter the escapement, either as to the locking faces or the impulse faces, which may be straight or curved.

Robin Escapement.

Old and new patterns.

887.—Fig. 14, plate X., represents the escapement invented in 1791 by Robin* and bearing his name.

The detent has two locking faces *m* and *n*. The unlocking is effected by the action of a small pallet, *b*, engaging in a fork at the end of a detent, and the impulse, which only occurs at

* Robert Robin, a clever French watchmaker, was born in 1742 and died in 1799. In addition to inventing several ingenious devices, he made some of the large turret clocks of our public buildings. The design and execution of his works are, as a rule, remarkably good.

every second vibration, results from the pressure of a tooth, *a* for example, against the face *c* of a notch in the roller.

The only action during the dumb vibration is to unlock the tooth resting against the face *n*, and to allow it to advance to the position *a* in fig. 14. This is preparatory to its giving the impulse during the return vibration.

A guard-piece or dart, not shown in the figure since it is covered by the fork, is, at the moment of unlocking, exactly opposite a notch cut in a roller which is carried on the balance-staff; the detent is thus prevented from overbanking.

888.—This escapement was much thought of on its first introduction; it was, however, very soon abandoned. We shall presently explain the reason for this.

Attempts were made a few years ago to restore it to favour and to improve its form. This new design certainly has some slight advantages over the original; but, as it does not avoid the primary fault of the escapement, there is little likelihood of the efforts for its introduction being crowned with success.

Fig. 54 shows one of the modern forms of the Robin escapement.

The roller (shown dotted) and the fork, *b*, have the same forms and perform the same functions as in the lever escapement.

A smaller roller *a*, carried on the balance-staff, is provided with a ruby impulse-pallet *c*. The escape-wheel is represented locked against the pallet *d*. Assume the balance-staff carrying the roller *a* to turn towards the left; the pallet *c* will pass in front of two teeth without engaging with either. But, when the balance returns towards the right, the ruby-pin *n* will enter the notch in the fork and, impelling it towards the right, will release the tooth *h*. The tooth *r* now falls against the locking face of the pallet *f*, and the wheel rotates through the distance that separates *r* from *f*. The tooth *p* is thus brought somewhat forward, as indicated by the dotted lines.

In the return vibration, when the balance is brought back by the balance-spring, a second unlocking will take place and, the wheel being free to move, a tooth will fall on to the pallet *c*, and communicate an impulse to the balance, which terminates with the tooth *q* reaching the arm *d*.

889.—Other Robin escapements have also been made resembling fig. 11, plate X., with the auxiliary spring removed; the detent terminating at its upper end in a fork and the recover-

ing pallet *s* being inclined in the reverse direction, so as to form a locking face.

Fig. 54.

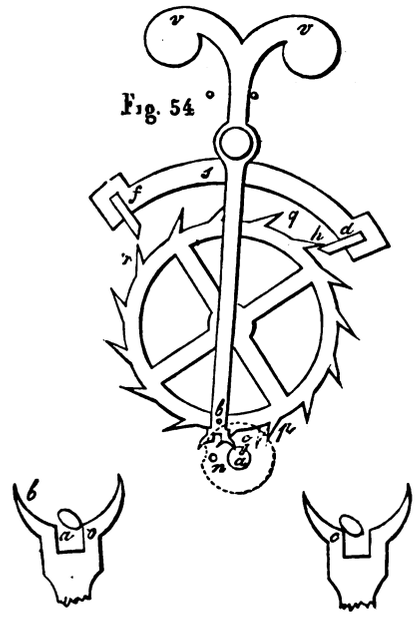

Fig. 55.

Fig. 56.

This latter construction will be seen to only differ from that previously described in the number of teeth that are included between the impulse and locking pallets; the action is identically the same as in the first escapement in 1791, and neither of these two varieties can be considered an improvement on it; in both the main causes of irregularity of the Robin escapement are retained as we shall proceed to show.

They require just as much care in pitching as a pivoted detent escapement; especially when formed as shown in fig. 14, plate X., where the total displacement of *n* is very slight. As regards this point the form given above possesses advantages; but the length of the fork-arm and the necessity for banking pins near the poise end may give rise, in the long arcs, to marked effects of flexure. It is to these causes, especially the one just mentioned (to which we first drew attention in the *Revue Chronométrique* in 1855) that we attribute the final abandonment of the Robin escapements, notwithstanding that several of them had given very excellent rates.

When the ruby-pin effects the unlocking of a tooth in the lever escapement, this tooth presses on the inclined plane of the

pallet-arm, and immediately the side *o* of the lever notch (fig. 55, page 519) engages with the ruby-pin and impels it forward. Since the lever alone is then exerting a force, its banking may be set more or less distant, according to the greater or less inclination of the impulse planes; this fact is taken advantage of to secure a sufficient interval of safety between the ruby-pin and the horn *b a*, and, besides ensuring perfect safety for the entrance and exit of the ruby-pin to and from the notch, every contact that would be detrimental is avoided.

But such is not the case in the corresponding period of action of a Robin escapement; it is not the fork that leads the ruby-pin, but this latter leads the fork, pressing against the side *c* (fig. 56). It necessarily follows, then, that unless there is a considerable amount of draw holding the detent back, the ruby-pin will catch against the corner of the horn both on entering and leaving the notch. If we take account of the oil, and of the fact that very many detents quiver and rebound on falling against the banking, there will be no difficulty in recognizing the source of numberless irregularities, and it will be evident why the Robin was abandoned, and with what good reason the lever has been preferred.

Breguet's Escapement with natural lifts.

890.—Most of the escapements in use at the beginning of this century were inconvenient because they were liable either to setting or to tripping or to rapid wear, mainly owing to the intensity of the friction at certain points for which jewels had not then been introduced. Doubtless struck by these inconveniences, Breguet sought to avoid them by the introduction of his novel form of escapement with what he termed *natural lifts;* an expression that has reference to the fact that two lever arms are in contact during the lift, and then travel in the same direction through the application of a force almost perpendicular to the line of centres.

891.—The following is a description of this escapement.

The last wheel of the ordinary train, *u d* (fig. 10, plate X.), engages with another smaller wheel *o v;* on the axes of these two wheels are carried two escape-wheels, B of six teeth and A of three teeth.

The successive lockings of the wheels B and A take place against *z*, which moves on the same axis as the fork-arm. This latter is identical with that of a lever escapement.

The action of the entire mechanism will now be easily understood. A turn of the key will set the four wheels in motion. The tooth *g*, engaging with the impulse pallet *j*, leads it to *h*, giving an impulse to the balance. The piece *z*, being impelled towards the right, is then met by the tooth *n* which is locked. The entire system of four wheels is now stationary.

On the return of the balance, the fork, being carried towards the left, releases the wheel B, and the tooth *r* (which is now at *i*) engages with the second impulse lever until it reaches the point *n*; at this moment the tooth *g*, having started from *h*, is locked by *z*, as this arm has been restored to its first locking position.

The piece *z* is movable on the staff of the fork-arm but held in a line with it by two side springs *p*, *p'*. The object of this arrangement is to prevent *z* from coming in contact with the back of a tooth, as it might do when the mainspring is run down, in the case of either wheel, A or B, being forced backwards by an impulse arm. With the device indicated, the piece *z* will be moved sufficiently to one side to allow the tooth to pass, and will at once return to its normal position.

892.—Notwithstanding that the conditions which Breguet seems to have laid down are very perfectly satisfied in this escapement, ingenious and carefully worked out like all the inventions of this celebrated horologist, it does not give good results although it be very well made; but it has served a purpose in proving indisputably that to increase the inertia of the several mobiles and to multiply the points of contact is the surest mode of preventing uniformity, in an action so rapid and so nearly instantaneous as that of an escapement. It has also some value from an historical point of view, and for that reason we have reproduced it here.

Breguet made some callipers in which the two escape-wheels as well as the two wheels attached to them had the same diameters.

Mac Dowall's Single Pin Escapement.

893.—This escapement is nothing more than a modification of that exhibited by M. Deshays (**1022**) at the Exhibition of 1827. It attracted considerable attention at the time and many clock and watchmakers thought extremely well of it, for the

lockings took place under very favourable conditions, and the method of applying the impulse presented a much more complete solution of the problem of natural lifts than did the arrangement suggested by Breguet.

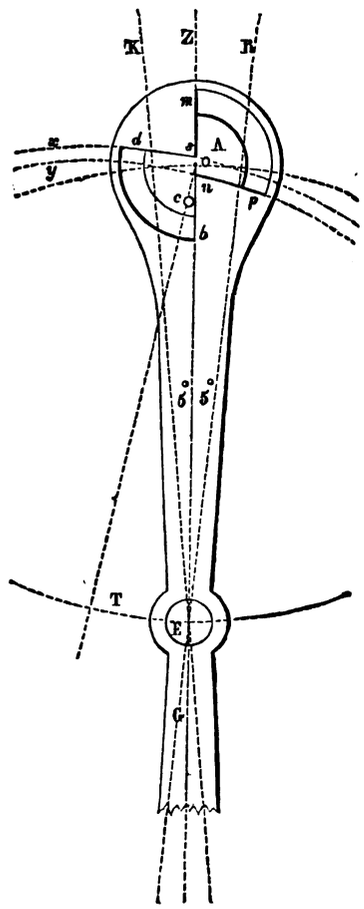

Fig. 57.

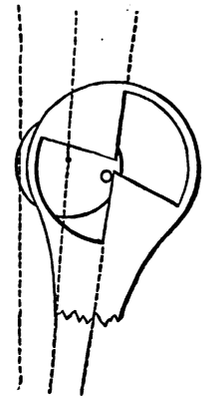

Fig. 58.

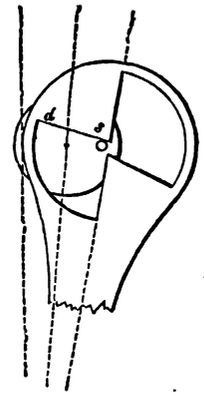

Fig. 59.

The escape-wheel is replaced by an ordinary plain wheel (of about 90 teeth), engaging with a pinion of 6 leaves, and on the axis of this pinion a small circular disc or roller is carried (A, fig. 57). This roller supports a pin *c*, projecting from its face. The roller in rotating forces the pin-shaped lever A G (which takes the place of the lever in an ordinary lever escapement) to the right or left according as the pin is engaged with the face *n b* or *s m* in the head of the lever. Assume the pin to have just acted on the face *n b* and thus to have accomplished a half-lift (figs. 57 and 58); it then falls against the circular arc *d s* (fig. 59),

and is locked. It continues thus until the balance, having received an impulse from the fork at G (formed precisely as in the lever escapement), is brought back by the balance-spring and effects the unlocking. The pin, coming in contact with the radius *s m* (fig. 57), forces it towards the left and, escaping from it, falls on to the circular locking face *n p*; while the balance, being entirely detached and having received the impulse necessary for the maintenance of its movement, completes its vibration. Its return occasions a fresh unlocking, etc., and so on through all succeeding vibrations.

The following is the mode of forming the head of the lever:

Through the centre of movement, E (fig. 57), and the centre of the roller, *o*, draw the line Z *o* E; then draw the two lines E K, E R, forming with the first the lifting angles Z E K, Z E R. These lines give the position of the pin in the roller, and therefore the size of the roller.

The piece of metal for forming the lever is carefully trimmed and made smooth and flat, and the line Z E drawn on it; drill the holes E and *o* on this line at a suitable distance. Taking the point *o* as a centre, draw the circular arc T E. Mark off the angle T *o* E of 12° (to represent the draw on the locking faces), and from the point T as a centre trace out the locking faces *x d s*, *y n p*. The interval between these arcs is the diameter of the pin, allowing the requisite freedom. When thus far complete the piece may be roughed out with a file.

The lever is usually made of gold and will work without oil.

894.—We have indicated the recommendations of this mechanism. Here are the faults that characterize it. It renders an additional mobile necessary, and this mobile, the energy of whose movement is inferior to that of an ordinary lever escape-wheel, must move with considerable velocity; this excessive velocity, as well as the smallness of the force producing it, causes the roller in time to become sluggish in its movements, because it is directly influenced by the oil and by every change in its condition.

This prejudicial effect, which interferes with the timing, cannot be avoided except by employing a very heavy balance: but we are thus on the horns of a dilemma; there must either be sluggishness in course of time or an excessive motive force. For this reason the very rare cases in which this escapement has been successful have not sufficed to secure its adoption.

Another Pin Escapement—sometimes termed the 7-tooth Lever Escapement.

895.—We do not know who was the inventor of this escapement, which is represented in fig. 13, plate X. It has been several times experimented upon in Geneva and, as one would anticipate from a theoretical study of its arrangement, it has never given very good results.

It is no more than a variety of the form last described and is characterized by nearly all its faults, in addition to being less certain in its action.

For when the unlocking takes place through the action of the fork, which is similar in every respect to that of the ordinary lever, the pallet *f* must have moved sufficiently towards the left to ensure that the tooth engaging with it is pitched deep enough to secure its proper action. The tooth *a* having been released and the first lift completed, a fresh locking takes place against the tongue *d*, at the conclusion of which the tooth *c* occupies the position *i*, the pallet *b* being also in the same position. These two projections must pass very near to each other, moving in opposite directions.

At the second unlocking, the pallet *i* is impelled backwards in close proximity to the tooth shown dotted, so as to be sufficiently in advance of this tooth at the instant of unlocking to receive an impulse from it.

All these actions involve great uncertainty; the intervals of safety must be reduced almost indefinitely so as to secure a deep enough pitching, and, without very great precision in the working of the several parts (only to be secured by most accurate workmanship in the escape-wheel and the head of the lever), there is a constant risk of catching. Moreover, if we take account of the thickening of oil, so sensible when the pivots fit exactly in their holes and the wheel is required to move with considerable velocity, it will be evident that we can, as already stated, predict from a theoretical study of the escapement, aided by a simple sketch, that it is far inferior to the ordinary lever.

Gontard Escapement.

896.—It consists of a detent *b c* (fig. 15, plate X.), moving on pivots and provided with a locking pallet at *b* and a recovering pallet at *d*. A very light gold finger *i d*, moving on the detent staff, is held against a banking *i* by the action of a flat spiral spring fixed at *c* beneath the detent. Otherwise the escapement

is similar to that of an ordinary chronometer with the roller reduced.

When the balance turns from right to left, the impulse pallet *e* passes close to two teeth of the escape-wheel, allowing a slight interval of safety without any contact, whilst the discharging pallet *f* meets the extremity of the finger *i d*. This finger yields, and, after moving a little backwards, escapes from the pallet and returns to its initial position, being brought back by the spiral spring. Nothing further occurs during this first vibration.

On the return of the balance, the discharging pallet again presses against the finger; but, since its farther extremity is now resting against the pin *i*, the detent is raised at *b* and the wheel released; one of its teeth engages with the pallet *e* and gives an impulse to the balance.

The detent is brought back to its position of rest, towards the conclusion of the lead, by the tooth *s* forcing the recovering pallet *d* backwards.

This escapement has been principally employed in small portable or carriage clocks, where its inventor substituted it for the ordinary detent escapement, as that is not satisfactory for ordinary purposes. It has the advantage (we are here not referring to the chronometers used for scientific observations) of being more easily adjusted by a watchmaker of average intelligence, as well as of possessing a smaller escape-wheel and a heavier balance. In the Gontard escapement, the only consequence of an accidental unlocking will be that a tooth passes too quickly.

We are afraid that it would be characterized by a rebounding of the detent and a rapid falling off in the crossings, with adhesion and perhaps even actual fixing of the collet that carries the finger, in case of oil reaching it; whereas it should work very freely on its axis and be kept dry. Many years' experience seem to show that when carefully made it gives good results; M. H. Jacot, one of our best makers of carriage clocks, has assured us of this fact.

Composite detached escapements.

Tavan's Escapements.

897.—Several watchmakers have made composite detached escapements; for instance, in part lever and in part Robin.

The lever-formed detent has two resting surfaces; one terminates with an inclined plane which receives an impulse from

the wheel to transmit it to the balance, and the other surface terminates in a point so that the wheel, immediately on the unlocking, acts directly on the pallet in the roller, and a second impulse is thus applied to the balance.

Such an escapement is inferior to the lever because it does not offer the same facilities of manufacture, and because it is characterized by at least half of the faults of Robin's form, even assuming that it is not liable to setting on the pointed resting piece of the detent.

898.—Of all composite detached escapements we believe the one invented by Tavan in 1805 and known as the *crab-claw* escapement to be the best. It has the two resting surfaces of Robin, but they are made of a circular form and the two pallets come near together, almost as in the pin escapement. The impulse is transmitted by means of a hooked pallet and under very favourable conditions, the friction being almost entirely disengaging.

We do not give a drawing of this elegant and ingenious escapement because it may be found described in almost every modern work on horology, and because all we can say of it at the present day, with advantage, is to point out its defects;—of two vibrations, one is dumb; the action of the unlocking pin in the fork is characterized by the same fault as in the case of Robin's escapement; lastly, it is more difficult to make and set in adjustment than a lever, and, except in the mode of application of the lift, is inferior to it from every point of view.

The same maker's escapement known as the *Echappement brisé et à surprise* is only a modified form of that just discussed. One of the resting faces is separate from the detent and supported between pivots, rendering it possible to materially diminish the drop; but there is no advantage secured by doing so because the number of contacts is increased, there being an additional staff and pivots, and the difficulties of construction are proportionately greater.

When we remember the period at which they were designed, we cannot help seeing great merit in many old forms of escapement, but they should not be employed in modern horology without a clear knowledge of their shortcomings.

Constant force or Remontoir escapements.

The constant force escapements for watches are also employed in clocks, and a few details relating to them will be found in article **1489**.

ESCAPEMENTS
OF CLOCKS AND TIMEPIECES.

PRELIMINARY.

Dead-beat and Recoil Escapements.

899.—The watch escapements already discussed are occasionally used in stationary clocks; but we shall not consider them here.

The escapements of timepieces and turret clocks that are controlled by a pendulum may be classified under four heads as in the case of watches, according as they are recoil, dead-beat, detached or constant force escapements. We are accustomed to this classification and shall retain it so far as the subject permits.

The construction of clock escapements, whether detached or undetached, must conform to the laws already enunciated in our Theory, of course omitting all that concerns the balance-spring; in describing these several escapements, then, it will only be necessary for us to give references to the paragraphs in which the principles are demonstrated.

The construction of recoil escapements must be governed by these same principles equally with others, and yet our manufacturers have not hitherto made it depend on any theoretical basis. All that has been published on the subject resolves itself into this: recoil accelerates the long arcs of oscillation (which are generally caused by an excessive motive force); it can, then, be taken advantage of as a means of increasing the uniformity in the rates of ordinary timekeepers. It will be evident that this is no more than a fact deduced from experience, a sort of makeshift that has been insufficiently studied, which watchmakers accepted although they thought light of it.

The natural consequence was that this class of escapements have been left more or less to chance and been made by inferior workmen. Excessive recoil has often been found necessary as a set off against the erroneous proportions of the mechanism as a whole, although it is a well-known fact that too much recoil always gives rise to rapid wear of the acting surfaces.

It has thus happened that, seeing the results obtained were usually very inferior, many watchmakers have unhesitatingly condemned these escapements without exception, maintaining

that the only escapements experience has shown to be suited to the best clocks are those with concentric locking faces.

This prohibition was a mistake and it has done much to impede the progress of the escapements now under consideration. Recoil endows the anchor with novel properties, both good and bad; the first of these can be taken advantage of in a given case or even in a good many, but not in all; and it is important, before finally proscribing recoil in every form, to thoroughly study it; especially with a view to ascertain whether it is possible to dispense with it in certain of the products of modern industry.

This is the whole question, for the recoil escapement exists in name only and not in fact; it is no more than an imperfectly dead-beat escapement, made imperfect intentionally.

The Recoil Escapement is merely a dead-beat escapement that has been unavoidably modified.

900.—The frictional rest escapement secures the greatest regularity in the going of a clock to which it is applied, when the locking face is of but slight extent and the proportions between the escapement arms, the height of impulse plane, virtual length of the pendulum, weight of the bob (as compared with the pressure that the acting surfaces can support), etc., have been duly adjusted.

These proportions can be investigated and arrived at without much difficulty, if not by calculation at any rate by trials conducted with intelligence, in the case of a large size Graham's anchor escapement; but the same cannot be said of the small anchors used in modern timepieces. For with them the play of the pivots is relatively considerable, whereas the surfaces of the inclined planes have been so far reduced that the exact form needed cannot be given to them; thus the loss of time and the drops become very great in comparison, and occasion great waste of force; lastly, the points at which the locking and the impulse should, theoretically, occur, in order to render the escapement but little sensitive to differences in the motive force, would be so near the centre of the anchor that it is practically impossible to get near enough to them.

But, such being the case, the dead-beat escapement, although doubtless the best *theoretically*, is *impracticable;* at least it requires in its construction the most minute care and a degree of precision that is incompatible with the demands of a very large industry.

901.—In a word, then, the dead-beat escapement, if not perfectly made in accordance with the requirements of theory, is unsatisfactory as a means of neutralizing irregularities in the force that maintains the movement of the moderator. It is then neither good for regulators, nor even for ordinary timekeepers; for in the latter case it is subject to the unequal action of a mainspring.

This fact is demonstrated beyond question, and we may observe that the introduction of a novel element, recoil, into this second class, which require less absolute accuracy than the first, would materially improve them; for we should thereby confer on these dead-beat escapements (which are nearly always found to be unsatisfactory and bad for the reasons already given) the power of correcting for variations in the motive force that they stand so much in need of; these escapements will then have a valuable characteristic, all the more so because its existence is in no way inconvenient, providing the recoil is not excessive so as to cause wear of the acting surfaces.

Properties of an escapement that are due to the existence of recoil.

The recoil must not be excessive.

902.—Let us consider an escapement with concentric locking faces as shown in fig. 4, plate XII., and we will assume that the extremity of the escapement arm is at *a*, the point that satisfies the demands of theory; in other words the length *b a* of the arm renders the escapement very approximately insensible to variations in the motive force. It results from the principles we have laid down in the Introduction to the study of escapements and in the several propositions, that when the arm is less than *b a* the clock will gain with the maximum and lose with the minimum motive force, and with a longer arm the reverse is the case.

903.—Now vary the shape of the anchor, first giving the locking face the form *n j* and afterwards that shown at *n l*, and let us consider what changes these alterations in the curved faces will introduce in the direction and resolution of the force (**228**).

The direction of the force, which, in the first position of the locking face, was along the tangent, *b a*, will be *c f* when the tooth is locked at *c* against a somewhat longer arm; the resistance opposed at the locking face increases with the depth to which the pallet engages with the teeth, that is to say with the extent of the arc of oscillation of the pendulum. This resistance, being

opposed to the motion of the pendulum, retards it, but the tendency is to hasten its return; the long arcs thus become more and more rapid in comparison with the short arcs, in proportion as the curve *n j* is more pronounced.

But the converse is the case with the curve *n l*. The gradual reduction in the length of the escapement arm and the fact that the direction of the force is below the point *b*, accelerate the short arcs, and, on the return of the pendulum, the tooth as it were holds the anchor in check. Hence it is evident that, as the arcs of oscillation become greater, they also become slower.

904.—The effects of such a curve as *n j*, which produces recoil during the *ascending* half of the oscillation, a recoil that may be called disengaging, have very frequently been observed in practice. As to those which, like *n l*, give recoil during the *descending* half of the oscillation, that is an engaging recoil, they have never been fully studied, since they were regarded as being of no possible use. At the same time this double recoil ought to be carefully investigated, since the two forms may occasionally coexist in an escapement in which the tooth rests against a straight locking face.

Experimental Facts.—Dent made some trials of the influence of engaging recoil; the loss on the rate that accompanied an increase in the motive force was very marked.

In the Memoirs of the Academy of Sciences two clocks with anchor escapements are referred to, in one of which any increase in the motive force occasioned a loss and in the other a gain, a circumstance which must have been due to differences in these curves, either intentional or occasioned by careless workmanship.

After these observations had been made, Romilly informs us that his father discovered experimentally, and Saurin seems to have demonstrated analytically, that the curve of the pallet in an anchor escapement should be very approximately the envelope of a circle in order that the escapement may itself counteract any irregularities in the motive force.

905.—In conclusion, then, it is important to remember that by the adoption of certain excentric curves results can be obtained that are the direct converse of those brought about by varying the length of the escapement arms with concentric lockings; and we thus are enabled to control the various elements of regularity and isochronism, employing them either in combination or apart according to circumstances (**921**).

906.—An escapement arm with concentric locking that is theoretically too short and requires the use of a curve analogous to *n l* is occasionally met with, but so rarely that the mere indication that it is possible appears to suffice.

907.—An escapement arm that is too long is very frequently to be found in modern clocks; and, either in consequence of the practical impossibility there is in securing the exact dimension needed, or of errors or difficulties in the manipulation, all the dead-beat escapements of commerce or very nearly all are influenced by variations in the motive force.

In this consists their fault; and this is the one point to which a remedy should be applied, that remedy being recoil.

908.—Before proceeding with this question we would observe that although moderate recoil is exceedingly valuable, in combination with a suitable length of escapement arm, as a means of procuring uniformity in the rate in the case of ordinary clocks, it increases the pressure on the locking face and must be carefully avoided in regulators. They are always the work of an accomplished maker and should be provided with an anchor that has concentric lockings; for this can be made accurately to accord with the demands of theory, or at any rate we can so nearly realize this condition as to secure all the advantages that the escapement offers.

The same may be said of turret clocks. In them the disadvantage of an excessive pressure is frequently met with, and any recoil would increase it.

While drawing attention to the benefit to be derived from the existence of moderate recoil in ordinary clocks, it is right to observe that if it exceeds a certain amount it can only secure a temporary uniformity due to an equilibrium existing between two faults, an excessive movement being balanced by an excessive pressure; this soon causes wear of the acting surfaces, when everything becomes uncertain.

In repairing clocks, when the workman cannot always take his choice of the method to be followed or replace pieces that are badly formed, a recoil that corresponds to the faults he has to correct will often be of the greatest service; but he must remember that perfect contacts can only exist when this recoil is of moderate extent and the teeth of the wheel not too narrow (**41**), slightly beaded crosswise and smoothed, say, with soft charcoal.

EXPERIMENTS OF F. BERTHOUD ON RECOIL AND DEAD-BEAT ESCAPEMENTS.

909.—F. Berthoud determined experimentally the effects produced by recoil and dead-beat escapements, and the alterations to which they give rise in the movement of a detached pendulum. It is important to be acquainted with the results of these experiments, although the very positive conclusions that their author drew from them are disfigured by serious errors. It could not be otherwise, since he started from a wrong assumption, basing all his arguments on one fact, which he constantly confirmed, namely that with a dead-beat escapement there is always a loss on the rate when the motive force is increased. He ignored that property of the lever which is fixed by the point we term the *theoretical point;* a property already, we hope, clearly explained in previous articles (**100, 239**, etc.).

910.—The instrument employed by Berthoud for these researches comprised three moving parts: the first was driven by a descending weight attached to a cord round a cylinder on the axis, the second was the escape-wheel, and the third the anchor supported on its axis and connecting the train and pendulum.

The length of the pallet arms could be increased or diminished at will.

"On the pallet-staff," says Berthoud, "I fitted three forms of anchor so made that, in conjunction with the same escape-wheel, one was dead-beat, the second caused considerable recoil, and the third gave an isochronal escapement; they were made so as to give the same lifting arcs.

"The escapement with excessive recoil caused the wheel to move backwards through an angular space equal to that measured by the inclined plane of the pallet.

"The isochronal escapement gave a recoil of the tooth equal to one-quarter this distance."

Before commencing the main experiments Berthoud set the pendulum to vibrate freely, and adjusted its movement to correspond with that of a seconds clock. "When this is done," he says, "we shall be certain, after one of the escapements above referred to has been brought to act on it, that any variations in the rate of the machine are due to this escapement."

EXPERIMENTS WITH THE DEAD-BEAT ESCAPEMENT.

911.—"I applied a dead-beat escapement with a lifting arc of 5·5°, a motive force of 30·59 grms. (about 1 oz. Troy) and

an arc of oscillation of 8°. In an hour the clock lost 30 seconds, or at the rate of 12 minutes per 24 hours.

"Without altering the adjustment I doubled the motive force; the arc of oscillation of the pendulum increased to 12°, and in an hour the clock lost 35 seconds, or 14 minutes per 24 hours.

"Employing the same escapement with three times the initial motive force, the pendulum described an arc of 14°, and there was a loss of 37 seconds in an hour, or 14 minutes 48 seconds in a day.

EXPERIMENTS WITH PALLETS CAUSING EXCESSIVE RECOIL.

912.—"On the crutch axis I fitted an anchor whose pallets occasioned very considerable recoil; the lifting arc was 5·5°; arc of oscillation, 8°; motive force as before, about 1 Troy ounce. In one hour there was a loss of 15 seconds; in 24 hours 6 minutes.

"With the same escapement; the motive force doubled; lifting angle the same; arc of oscillation, 10°; loss in one hour, 6 seconds; in 24 hours, 2 minutes 24 seconds.

EXPERIMENTS WITH PALLETS GIVING MODERATE RECOIL (ISOCHRONAL).

913.—"The anchor with pallets of moderate recoil was next fitted on to the axis: lifting arc 5·5°; arc of oscillation, 8°; motive force, about 1 Troy ounce as before; in one hour there was a loss of 27 seconds, at the rate of 10 minutes 48 seconds per 24 hours.

"The escapement and its lifting arc remaining unchanged, the motive force was doubled; arc of oscillation, 12°; loss 27 seconds per hour, or 10 minutes 48 seconds per day.

"Same escapement and lifting arc; motive force trebled; arc of oscillation 14·5°; loss in 1 hour, 27 seconds, and, in 1 day, 10 minutes 48 seconds.

"These experiments suffice to show the manner in which the oscillations of a pendulum are modified according to the character of escapement."

914.—To sum up, then, under the conditions laid down by Berthoud, increase in the motive force occasions a gradually increasing loss on the rate with a dead-beat escapement;

There is a gradually diminishing loss with an escapement of considerable recoil;

And, lastly, the rate remains invariable when the pallets have moderate recoil.

"This last form of escapement renders the oscillations of the pendulum isochronous notwithstanding that both the motive force and the arcs described are variable."

915.—Fétil observes that Berthoud has concluded too much from his experiments and that, if he had caused the motive force and the weight of the pendulum to vary in other proportions, he would probably have deduced other results which were more favourable to the dead-beat escapement, and therefore less to the advantage of the recoil.

This remark is legitimate; had it occurred to Berthoud himself he certainly would never have written, "The dead-beat escapement, while being the worst for clocks actuated by a spring is assuredly not the best for watches," and he would unquestionably have had some idea as to the proportions best suited to the dead-beat escapement.

And yet he well knew, as is proved by the following extract from his work, what differences are occasioned in the rate by varying the motive force and the weight of pendulum concurrently. The latter changes modify the arcs of oscillation, and therefore the momentum of the regulator, in a ratio that nearly always differs very considerably from the rate of increase or decrease in the motive force.

OBSERVATIONS MAINLY CONCERNING THE WEIGHT OF THE PENDULUM.

916.—"In a clock provided with a dead-beat escapement, the more we diminish the weight of the pendulum bob and the arc of oscillation, the more will the clock lose on increasing the motive force. If a recoil escapement be used, while the bob is too light as compared with the motive force, it will be found that on increasing this force, the oscillations of the pendulum are more rapid. . . . From these observations it follows that the character of any given escapement changes when the momentum of the pendulum as compared with the motive force is modified. Thus the curved face that effects the recoil will not be the same in different types of clock; it will be essential to take account of the greater or less weight of the pendulum bob, the motive force, arc of oscillation, etc."

Berthoud's instructions for making his isochronal pallets will be given subsequently; we would only observe, as bearing on the last remark, that every change in the weight of the bob must involve a corresponding alteration in the uniform action of the escapement, either at once or after a definite interval of

time, through one of its principal conditions having been modified: the ratio of the length of the pallet-arm to that of the pendulum (assuming its virtual length, that is approximately the length of the corresponding simple pendulum, to remain invariable) remains practically the same, but the resistance opposed to the impulse by this pendulum has been altered. We shall recur to this subject.

PRINCIPLES THAT REGULATE THE CONSTRUCTION OF DEAD-BEAT AND RECOIL CLOCK ESCAPEMENTS.

917.—We consider it to be satisfactorily demonstrated in our new Theory that it is possible to fix upon a proportion between the length of pallet-arm and of pendulum that will render the escapement insensible to variations in the motive force or very nearly so.

But have we proceeded to conclude, as some watchmakers seem to think, that all escapement arms should have precisely this length?

Certainly not; for not only would such a conclusion be beyond what we have actually proved; it would be absolutely false.

It is essential to take account of the greater pressure exerted on the pallets when their arms are short and to modify the theoretical length in accordance with our knowledge of:

(1) The inconvenience arising from this excess of pressure;

(2) The influence that it may have on the going of the escapement.

Relation of length of Escapement arms to pressure.

918.—Since friction is proportional to pressure (**38**), the friction on the pallets during the lift will be somewhat more harsh with short escapement arms; thus their faces will deteriorate more rapidly than with longer arms, all other proportions remaining the same.

It is necessary, moreover, to deviate more and more from the theoretical length, as the resistance of the materials to wear and distortion becomes less; whether for instance, tempered hard steel, hard steel or stones be used. The employment of highly polished stones enables us to approximate most nearly to it, except in such a case as that referred to in the following note.

Remark.—Even if the pallets are provided with jewels it is at times necessary to somewhat increase the length of the escapement arms so as to avoid excessive pressure; when this pressure

becomes such that the point of the tooth drives the oil away from the acting surface of the impulse pallet, as though it were a sharp blade drawn perpendicularly along a hard oiled surface, the friction is the same as between unoiled surfaces; for the oil is temporarily displaced, returning to its initial position after the passage of the blade.

The effect of this pressure on the rate varies with the length of the pallet-arm.

919.—We have considered the intensity of the pressure merely from the point of view of friction, having regard both to the preservation of the acting surfaces and the irregularities that their deterioration might involve; it remains for us to investigate the effect different degrees of pressure on the lifting faces may have on the timing (employing the same lifting angle and a definite velocity of rotation of the escape-wheel); attention has, we believe, never previously been drawn to this point.

The force exerted by the wheel against the impulse face (ignoring that which is due to the drop) is decomposed. We may consider it to be divided into two parts, more or less unequal, and of these one is converted into friction and the other is available for increasing the movement of the moderator (**238**).

Assuming a constant force to impel the wheel, the portion wasted in acting as a check on its velocity, or in other words in intensifying the friction, becomes greater as the weight of the bob is increased. On the other hand, when the bob is lighter, the pallet-arm will yield more readily on pressure being applied, so that the wheel acquires a greater velocity; the intensity of friction is reduced, and all the force saved by this diminution is added to that employed in moving the moderator; its arcs of oscillation increase but they at the same time become slower; the long arcs occupy a longer period than the short ones.

920.—Those of our readers that wish for a theoretical demonstration of these facts can easily devise such an one with the help of the Elements of Mechanics collected at the beginning of this work, or given in the course of our Theory of escapements; and they will thus be enabled to demonstrate the following curious and important propositions, dependent on the weight of the pendulum.

PROPOSITIONS.

A dead-beat escapement with concentric locking faces (the arms being of the theoretical length) will, if timed, give, with

the same virtual length of pendulum (**1016**) and an increased motive force:

—A *loss*, if the pendulum-bob be made lighter, and a *gain* on the rate if the length of pallet-arm be diminished while the bob remains unaltered;

—A *gain*, if the weight of the pendulum-bob be increased, and a *loss* if the pallet-arm be lengthened without changing the weight of the bob.

These results are very remarkable; there are four distinct phenomena, two of which are characterized by a loss whereas the other pair involve a gain, and we thus possess two modes of securing the maximum of uniformity.

921.—With a dead-beat escapement, the long arcs lose on the shorter ones.

The converse is the case with certain recoil escapements.

It therefore follows that if the escapement discussed in article **920** were converted into a recoil escapement, it would give, on increasing the motive force:

—*Loss*, if the bob be made heavier;

—*Gain*, in the contrary case.

In other words precisely the converse of what occurs with a dead-beat escapement.

Enough has been said to show how far advantage can be taken of these important properties by setting them one against the other, or combining them with a lifting angle of the most suitable extent; a combination which should above all be effective in rendering the escapement practically insensible to variations in the motive force, whether due to irregularity in the action of the motor or to thickening of oil.

The Lifting arc and the Supplementary arc.

Proportion between the lifting arc and the velocity of movement of the pendulum.

922.—In order that the foregoing considerations may be true, it is absolutely necessary that the lift be effected with uniformity, that is to say in the manner secured by the plane we have termed the mean incline (**224, 255**). If any loss of time, slipping or jerking, etc., occur during the lift, the above phenomena may not be found to occur or they may even be reversed; such a fact would indicate some error in the design or faulty construction. The choice of lifting angle is, therefore, by no means a matter of indifference (**672**). It must, in the first place, be such that the action of the tooth against

the impulse plane is accomplished with the least possible resolution of the force applied, and in accordance with the conditions indicated in paragraph **272**.

923.—As to a law that would guide us in the selection of this lifting angle, in other words one that determines the angular measure of the impulse that should be communicated to the escapement arms, we look for one in vain in the best works on clockmaking. They are silent on this subject, and usually content themselves with saying that the lift should be "suitably proportioned," which is self-evident, and might just as well have remained unsaid; and they attempt to make up for the absence of any rules by enumerating the proportions that have been found satisfactory by makers of repute.

We are thus again face to face with a novel question. With a view to be concise in our explanations on this point, we will at once summarize them in the manner we have already several times adopted.

PROPOSITIONS.

For every pendulum (with a given motive force):

924.—First.—The number of oscillations in a given time depends on its virtual length (Theory of the pendulum);

925.—Second.—The angular velocity varies in a certain proportion, inversely with the mass;

926.—Third.—The lifting angle depends on this angular velocity.

To take an example:

927.—Consider a pendulum beating seconds and oscillating through an arc of 4° (*cj*, fig. 60), under the influence of a given motive force.

Retaining everything else, including the virtual length, etc., the same, double the weight of the bob.

From the law that *when a given force acts on different mobiles they move with velocities that are inversely as their masses*, it follows that the new pendulum will still beat seconds, but it will only have a path of about half the amount. The angular velocity will, then, be diminished in this proportion.*

But, owing to this reduction in the velocity with which the impulse arm is displaced, the escape-wheel is unable to acquire the requisite angular velocity, in other words a velocity such that

* We are here considering a particular case in order to make clear one aspect of the question; but the reader must not generalize from it, as he would then fall into serious errors (**929**).

the impulse it communicates is sensibly the same, except by being made to act against a plane of greater inclination; that is to say, unless the angle of the lifting face be diminished in a certain proportion according as the weight of the bob is increased (**930**).

928.—Let us revert to the first pendulum (A *c*, fig. 60), which described an oscillation of 4° in a second, and assume the bob to be transferred from *c* to *o*, that is to the point at which the pendulum would beat two oscillations in a second.

The mass remains the same but it is now set at the extremity of a lever-arm that has only one-quarter the length; it is, then, impelled by a force that is relatively four times as great, and, assuming everything else to remain the same, this mass should acquire in the same period, an angular velocity that is four times as great, in virtue of the law of the proportionality of force to velocity (**120**); this would give for the short pendulum under consideration

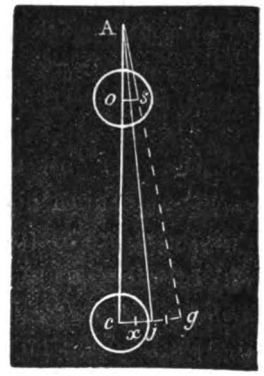

Fig. 60.

an angular movement of slightly less than 8° per half-second (the paths traversed in the same time, *c x* and *o s*, or *c j* and 2 *o s*, being equal).

The inertia of the escape-wheel prevents it from suddenly acquiring such a velocity as would ensure its following the pallet and pressing against it; the wheel, moreover, cannot maintain the motion of the pendulum unless the lifting angle is increased in the same proportion as the velocity of the pendulum bob is augmented; for otherwise the impulse arm would be deprived of the action of the wheel in consequence of the velocity with which it moves, and there would only be a slight impulse and possibly, even, none at all.

929.—The complete solution of the problems we have just considered in an elementary manner is very complicated, in-

volving the use of high mathematics; we have therefore been content to obtain approximations that are more or less near to the actual value; to explain merely such of the leading theoretical data as are needful as a guide to the experimental method; a method that will, at the present day, enable us to quickly arrive at definite results.

930.—In order to facilitate the explanation, we have assumed the lifting action to remain the same notwithstanding that the angular movement varied; but it is necessary to draw attention to a fact that must not be forgotten, namely that the energy of the lifting action depends on the rapidity of the angular movement, and, therefore, on the form of the surfaces which, by acting on each other, produce the lift (**272**).

931.—While being perfectly regular, the lift should depend on the velocity with which the free pendulum returns to its position of rest, when moved from the vertical through an angular distance equal to the arc of oscillation; and the motion must be accelerated without any sudden pressure. The regularity will be the better assured according as the constant and uniform action of gravity is less interfered with.

The supplementary arc.

932.—It seems so evident as not to require demonstration that the lifting arc and supplementary arc are correlated, and that their relative extent should vary:

(1) With every change in the motive force;

(2) With every alteration in the height of the impulse plane (depending on the interval between the teeth of the escape-wheel) or in its inclination (**604**).

In both cases the ratio between the energy of impulse and the friction on the locking faces may become more or less advantageous, but its influence will certainly alter and the nature of the change must be determined by experiment.

933.—Speaking generally, it is advisable that the supplementary arc be very limited (it is made about half the lifting arc, and usually even less), because it is more sensitive than the other portion of the oscillation to changes in the condition of oil, and an increase in its extent corresponds to a much more rapid increase in the impelling force.* When the motive force is

* This is the converse of what we have found to be true of watches. For that case is different since the action of gravity is eliminated and the balance-spring enables us to vary the velocity of the moderator and make its long and short arcs of vibration sensibly isochronal.

variable, any accidental excess in the supplementary arc may be prevented by giving a slight recoil, more especially during the movement beyond the normal supplementary arc, as is done by some makers of pin escapements.

General Practical Rule.

934.—We will terminate this summary, which will be found sufficient by a reader who is thoroughly acquainted with the principles set forth in the Introduction to the study of Escapements and in our several Propositions, by the following advice:

It is possible to determine whether the several parts constituting the escapement bear the best proportions to one another or approximate thereto, by observing the two following features, occasionally separate but very frequently existing together:

(1) The extent of the arcs of oscillation remains constant, or very nearly so, with a variable motive force; the mainspring being either fully wound up or nearly run down.

(2) The isochronism of the oscillations is maintained or only suffers very slight disturbance however the motive force be modified; a circumstance that may be verified by comparing the clock during equal times with a regulator beating seconds.

Equality in the extent of the arcs does not necessarily imply that they are isochronal. The isochronism is, moreover, more or less modified by the nature of the suspension, as we shall subsequently show (**979**).

ESCAPEMENTS OF CLOCKS AND TIMEPIECES.

Verge Escapement.

935.—The verge escapement is used at the present day in timepieces that are known as Comtoises and in them has given excellent results. It is also met with in many old clocks.

The recoil becomes less inconvenient as the opening of the pallets is reduced; it thus happens that in well-made Comté clocks, driven by a weight and having a long pendulum rod that possesses a certain degree of flexibility, the influence of the recoil is inappreciable.

Considering their facility of manufacture and moderate price, these clocks, of which nearly 100,000 are made annually, constitute an industry that deserves to be encouraged. Although little care is taken in their construction, they give results that are very satisfactory for ordinary purposes. They appeal, more-

over, to a class of purchasers that require a substantial mechanism which can be repaired at moderate cost.

936.—The maker who is desirous of possessing a thorough knowledge of the verge escapement should select, from the chapters between pages 54 and 93, whatever is applicable to clocks. Mainly in article **163** will he find the leading data, and they are corroborated or corrected (if necessary) by the following:

Average lifting arc, from 5° to 7°;

Supplementary arc, from 4° to 6° (2° to 3° beyond either side of the lifting arc); hence the total oscillation is between 10° and 15°;

Weight of pendulum, from 135 to 150 grms. (4 to 4½ ounces); length about 95 cm. (37 ins.).

Length of pallets, about 13 mm. (0·5 in.). Opening between 50° and 60°.

Driving weight (from 8 to 9 lbs. for an 8-day clock) increases with the period the clock goes. Avoid excessive pressure since it makes the rate irregular.

It must be observed that these experimental data are in no sense absolute; they are only approximate guides to serve as starting points in attempting to ascertain the most advantageous proportions.

937.—A maker desirous of securing the best possible arrangement of this escapement would do well to know, either from notes previously taken on the performance of clocks throughout a long period, or from a study of the experimental data collected by various practical men, by about how much the arc of oscillation diminishes in an average period, say of 4 or 6 years. Carefully conducted experiments, in which similar variations in the arc of oscillation are effected by modifying the motive force, while at the same time the lifting arc is altered within moderate limits, as well as the length of pallets and the weight of bob, would enable him to plan the whole satisfactorily.

It is especially important to remember that this escapement does not do with a very great motive force, a rigid pendulum rod or a heavy bob. When the motive force is considerable and the rod inflexible, the friction during recoil becomes very harsh; a heavy bob requires long pallets and these must be pitched shallow, the arc of oscillation is much reduced and very great accuracy in the action of the escapement becomes essential. The making of the escapement thus

becomes as delicate as that of one of the better class, while it can never by any possibility give such good results. It seems needless to insist on this point; the question is now finally settled.

RECOIL ANCHOR ESCAPEMENTS.

938.—The first anchor escapement appears to have been invented in 1680, by Clement, a London clockmaker. At any rate most authors give him the credit of it, but it has been claimed by Hooke.

In these early forms there was a recoil with each pallet, and the locking faces had certain curvatures which should, according to Saurin, be very approximately envelopes of circles (**904**). Their form has been somewhat modified: they are still made to produce a recoil with both pallets, but with the difference that the locking face of one is curved while that of the other is straight.

Lastly, escapements are also made which give a dead-beat on one pallet and recoil on the other, which from this circumstance are termed *half dead-beat* or *half-recoil* escapements.

We will first describe the isochronal escapement of F. Berthoud, already referred to in the account of his experiments (**913**).

Isochronal anchor escapement of Berthoud.

939.—First draw the arcs $p\ c$, $g\ r$, $d\ t$, and $f\ b$ (fig. 10, plate XII.) as if for an anchor with concentric locking faces. Then, in order to determine the curves that produce the requisite recoil, take the interval $s\ b$ (the width of the pallet) and mark it off three times along the arc $b\ f$. Through the points a, 3, draw the line $a\ h$. From the point 3 measure off the distance 3 h, equal to $s\ b$; from h as a centre with the distance $b\ a$ as radius, draw a circular arc, and from the centre b with the same radius draw another arc, cutting each other at the point n. This gives the centre for the locking face $b\ l$, which can now be drawn, of course using the same radius $b\ a$.

Proceed in the same manner to determine the points h' and n' for the inner face of the other pallet; the mode of procedure will be easily gathered from the figure.

It should be noticed that any change in the size of wheel or the number of teeth will render a fresh drawing necessary, etc. (**916**). This escapement has rarely been employed.

Ordinary recoil anchor escapement.

940.—Figures 3 and 5 of plate XII. give the two forms of recoil escapement that are most usually met with. The second of these is generally known as the small recoil anchor and the first is termed a gable anchor. Such escapements are only used

in the cheaper class of clocks for they can be made with ease by ordinary workmen. The first is employed in conjunction with either a silk or spring suspension, and the pendulum weighs from 15 to 30 grammes (0·5 to 1 ounce) with silk, and 30 to 70 grammes (1 to 2·3 ounces) with a spring; the second form has a spring suspension and a pendulum that is relatively heavier.

With a view to help those who may have occasion to remake or repair such forms of escapement, we will indicate the methods adopted in the Paris factories.

941.—*To draw the small recoil anchor.*—It is first necessary to ascertain the distance between the centres of the anchor and wheel if not already known, and the following is the mode of proceeding to determine it (fig. 5, plate XII.):

Draw on a brass plate the circumference of the wheel *c k g*. Mark off the distance *c g* between the points of teeth included within the pallets, increased by one; say 6 (**946**). Through the extremities of the radii L *c*, L *g*, draw two perpendiculars, and their intersection E will give the required centre. A fine hole should be drilled at it.

From the point E with a radius E *c* draw the circumferences *n c g b* and *a s d*, the latter accurately bisecting the space between two teeth. Draw the lines E Y, E Y′, passing through the points of the teeth that lie on the circle *n c g b*; then draw the lines E X, E X′, forming with the two first the lifting angle (usually 5° or 6°).

Through *a* and *c*, the points where the two lines limiting the lift intersect the circles, draw *c a i b* which determines the internal face of the disengaging pallet, and *c n* is its external face.

Similarly, by joining the points *g*, *s*, we obtain the impulse face of the engaging pallet; this should be continuous with the curved recoil face *g h*, which is to be drawn so that the angle of recoil, *m g h*, is equal to *v a i*. A slight excess of metal should be left, to be removed if, after trying the escapement in position, there is found to be no danger of a tooth touching the plane *a i*, during recoil on the other pallet.

The internal face *s d* of the engaging pallet may be straight or concave, for it is only necessary to avoid its coming in contact with a tooth.

942.—*Another mode of tracing the calliper.*—In practice a still more expeditious means of making the drawing may be resorted to. On the surface of a piece of steel from which the anchor is to be formed draw a straight line, and on this mark points

corresponding to *c* and *g*, indicating the distance from the first to the sixth tooth. The centre of the anchor will be set at a distance equal to one-third of *c g* above this line, and at equal distances from *c* and *g*.

After having drilled a hole at this centre, describe the circle *n c g m b*, taking E *c* as a radius: divide the upper semicircle into six equal parts and join 5 and *c*. Then draw the second circle, *v a s d*, accurately bisecting the interval between the points of two consecutive teeth. Mark off the distance *m h* equal to *v i* (many makers do not even measure it, but estimate the dimensions by the eye), and we now have all the necessary proportions, so that we can easily cut the anchor by their aid.

943.—*To make the anchor and set it in adjustment.*—The steel having been carefully prepared and the axis properly fitted in a central hole, the several lines that are necessary are drawn on the surface of the metal, and it is formed to the required shape but without removing quite all the metal along the line *s d*. The anchor is then made to engage with the escape-wheel in the depthing tool as is done in the case of watch escapements: when a tooth rests at *g* there should be the least possible freedom between the tooth *c* and the face *c n*. With a tooth resting at *a*, the fifth tooth to the right should not engage at all with the face *s d*, or it should only just do so, in order that it may be possible to make up for any possible distortion of the pallets in hardening; the anchor is now hardened, and the freedom at *s* is adjusted when smoothing the face *s d*; it is then finished, being tried on the depthing tool until found to be correct (**944**).

Those who do not possess a depthing tool of large dimensions can adopt the following plan, which is much used by escapement makers engaged on common work. It is not very exact, but may do in the absence of a better means.

The axis of the wheel is driven into a piece of cardboard, and, at distances approximately equal to the interval between the two centres, a number of holes are made of the same diameter as the axis of the anchor. Having set the wheel in position, the anchor is successively centred in different holes until one is found with which the action of the escapement is possible; then proceed as in verifying on the depthing tool, modifying the shape of the anchor as required.

Such a method may be resorted to in an emergency, but it would be better to mount the anchor (on its axis) in a sliding

frame so that it can be brought to and from the wheel by a definite amount.

In hardening, the anchor should be introduced into the liquid with its back downwards, as the pallets have been observed to open out least when this precaution is taken. All qualities of steel are not equally distorted; a fact which the escapement maker should bear in mind.

944.—OBSERVATIONS: *First.*—The two recoils should be equal, or, at least, they ought to occur in the manner that is most favourable to the movement of the pendulum; it follows, then, that a relation should exist between the inclination of *c a i* (fig. 5, plate XII.) and the curve *g h*, and that whenever one is modified the other should be altered in a corresponding manner. It is manifest that, under the conditions that actually occur, and with the methods commonly employed at the present day, the mere mechanical imperfections of the work would be sufficient to render all exact determinations of the most advantageous recoil useless. Whenever they do happen to be well proportioned it is accidental; for we must again point out that the exact equality of these recoils is of less importance than equality of the lifts, and, in this class of escapement, these are nearly always very unequal as regards the impulse they transmit to the pendulum.

945.—*Second.*—The lift here measures from 5° to 6°; the reader will remember that it depends on the length and weight of the pendulum. (See the chapter summarizing the experimental data, articles **1010** to **1012**.)

946.—*Third.*—In tracing the form of the anchor, we have taken as a basis of measurement the distance between the point of one tooth and that of the sixth from it; this datum is suited to the three and a quarter inch movement, provided with a pendulum measuring 8 inches in length and a wheel of 34 teeth about 0·8 inch in diameter.

If the wheel be larger or smaller, of a different number of teeth, with a shorter or longer pendulum, it will of course be understood that the number of teeth covered by the anchor must be altered, in order that a proper proportion may exist between the length of the pallet arms and pendulum (see the chapter commencing with article **1019**).

Recoil escapement with triangular or gable anchor.

947.—With this class of anchor, employed in conjunction with a spring suspension and a somewhat heavy pendulum (from

5 to 10 ounces), the friction on the recoil faces is considerable and they generally wear rapidly. The irregularity resulting from this friction, which gradually becomes more and more harsh and variable, is the more marked as the escapement arms are made longer.

Clockmakers have at the present day abandoned the gable anchor in favour of the English form or that known as the Brocot anchor. Nevertheless we will indicate the mode of procedure, as they are at times called upon to make it.

948.—*To draw the recoil gable anchor.*—Figure 3 of plate XII. shows the form most generally in use. The anchor usually covers 10 teeth, or, rather, 11, counting from the beginning to the end of the two impulses (for a wheel 0·8 inch in diameter, with 34 teeth and a pendulum measuring about 8 inches.—See **946**, 2nd paragraph).

Ascertain the distance between *d* the centre of the anchor and *m* that of the wheel (**941**). It is in the present case about 1·8 times the radius of the wheel. This amount is to be increased or diminished, according as it is desired to reduce or augment the extent of oscillation of the pendulum.

Knowing the distance between these centres, describe the circle *b g*, with such a radius that it passes through the points of one tooth, *a*, and of the eleventh from it, *o*. Then describe a second circle, *b′ g′*, passing exactly through the middle of the space between the tenth and eleventh teeth.

Draw lines indicating the lift as in the former case; this generally measures about 4° (**945**). The impulse faces are determined by the intersections of these lines with the circles *b g*, *b′ g′*.

The prolongation of *a c* above the circle *b′ g′* determines the recoil face of the disengaging pallet; and, in order to obtain an equal recoil on the engaging pallet, the impulse face must be continued by a curve described with the radius of the wheel, in such a direction that the angle *n o g* is equal to the angle *b a c* (**944**).

Clockmakers as a rule do not take so much trouble in the matter; they fix upon the curve by the eye without paying much attention to the equality of recoil (correcting for the distortion in hardening).

If the anchor opens too much in the hardening (**943**), it may be slightly closed by screwing the two arms together, the body being at the same time let down to a bluish-grey; but this must be done with care, and the body should have been previously blued and re-polished.

949.—*Details relating to a gable anchor escapement.*—The escapement in question maintained its rate well (for ordinary purposes), and did not show any signs of wear after going for several years; its mainspring was soft and pliable.

Diameter of the movement, 82 millimetres (3·23 inches); spring suspension.

Diameter of wheel, 20 mm. (0·8 ins.),—37 teeth.

Anchor covered 8 teeth; lift, 3°; supplementary arc, 1°.

Total arc, about 5°; varied slightly as the spring was more or less wound up.

Pendulum: Weight, 185 grammes (about 6.8 ounces).—Virtual length (**1016**), 164 mm. (6·45 ins.).

The length of escapement arm compared with

The weight of pendulum is :: 1 : 17,

(using French measures, the millimetre and gramme).

And with the virtual length :: 1 : 15.

English recoil anchor escapement.

950.—We are not aware why this form of anchor (fig. 11, plate XII.) is termed English, for we are informed that it is not employed in England.

It is nothing more than the old gable anchor shortened, so as to cover a less number of teeth; a change which renders necessary an alteration in the original form and a reduction in the weight of the pendulum.

This escapement is usually provided with a spring suspension, a pendulum weighing from $2\frac{1}{2}$ to 7 ounces (**1016**) and a mean lift of from 4° to 5° (**945**).

951.—*To draw the calliper of the English anchor.*—It is usual to proceed in the same manner as when drawing the recoil anchors already discussed (**941**) with the following modifications: the larger circle usually includes 9 teeth; after dividing the semicircle *a* 7 into 7 equal parts, the inclination of *a d* is given by joining the points *a*, 5; and the recoil face *i c* should project slightly beyond the line *x z*.

To conclude, this drawing is, like those which precede it, only intended as a means of facilitating the work of making the anchor of approximately correct form; it is always necessary, when exact workmanship is required, to verify the lifts, recoils, etc., on the depthing tool and in the manner explained for watch escapements (**943** to **949**).

952.—Many makers prefer that the face *i c z* be perfectly

straight, but they give no reason for this except that the labour is thereby reduced. This form, as usually made, diminishes the energy of the drop, but it at the same time increases the harshness of the recoil.

Rozé's recoil anchor escapement.

953.—The anchor of this escapement, which is made in accordance with the calculations of M. Rozé senior, differs materially from those hitherto considered, especially as regards the form of its impulse faces, which, instead of being plane, have a pre-determined curvature (fig. 7, plate XII.).

M. Rozé* has, with the assistance of his son, completed his work by proposing a calliper that enables us to form these curves with great facility; avoiding the difficulties that have brought about the failure of most attempts at making the impulse faces curved.

With a 30-tooth wheel the anchor covers $4\frac{1}{2}$ spaces between the tangents; that is to say, the points of six teeth counting from the beginning to the end of the double lift.

The radii of the locking faces o c, o c′ are equal.

954.—The Lift.—The lifting angle is 2°.

Datum.—The lift is progressive; that is to say, half the total movement of a tooth during a lift corresponds to $\frac{2}{3}$rds of a degree (40′) motion of the anchor, and the other half causes a movement of $1\frac{1}{3}$ degree (1° 20′).

This progression is intended to enable the motive force to overcome with greater ease the resistance due to the inertia of the train at the commencement of the lift, and to reduce the energy of the impact on the locking faces to a minimum.

We cannot here enter into the requisite theoretical considerations, as they would take us beyond our limits, and we will only say, in explanation of the mode in which the progression here referred to satisfies the required conditions, that attention should be paid to the variable velocity of the pendulum. The calculation allows for an angle of 0·5° to be added to the angular path of the wheel on account of the thickness of the tooth, inequalities, thickening of oil, etc.; and for another angle of 1°, added to the lift, in order to ensure certainty of action

* A. C. Rozé, who died in 1862 at the age of 50 years, was one of the very few eminent horologists that have realized the fact that theory and experiment must always be associated; and that, in our day, no true progress can be made without considerable theoretical knowledge. His labours, which were unfortunately interrupted by a sudden death when his talent was at its best, are considered in an article that was published in the *Revue Chronométrique*.

and proper pitching with the locking faces. This angle of 1°, in addition to the 2° of lift, gives 3° for the lifting arc of the pendulum. No account has, however, been taken of the distortion of the arms of the anchor that frequently occurs during hardening (**943**).

955.—We would observe that in forming the pallets as here explained, there is sometimes danger of the teeth being damaged after an exact relation has been established between the anchor and pendulum, especially as regards the engaging pallet. Such an accident is avoided by increasing the lift or by lengthening the pallet-arm so as to make the pressure less perpendicular to the tooth, or else by using a movable anchor that is held on its axis by stiff friction.

956.—The Recoil.—This escapement can, as is shown in fig. 7 (plate XII.), be made either dead-beat, *m* o′ *n*, or with recoil, *c*′ o c. We shall only here consider the second form.

Theoretical Datum.—The successive angles that measure the recoil of the wheel are proportional to the square of the amplitude of the oscillations; that is to say, they follow the same law of progression as the periods, with varying arcs of oscillation (see the article on the *Pendulum*).

Practical Datum.—If the pendulum made a supplementary arc of 22° on one side, the wheel would recoil through a distance equal to half the interval between two teeth.

These recoils differ from those advocated by Berthoud (**939**), for with his the increase in the recoil of the wheel for a given increase in the supplementary arc becomes gradually less as this arc becomes greater, whereas here the converse is the case.

957.—*Tables of the dimensions of this escapement.*—By employing a series of tables calculated by M. Rozé and giving the exact dimensions of the anchor when the number of teeth of the wheel is known, it becomes an easy matter to draw the calliper of the anchor and to make it accurately.

We give the table for a wheel of 30 teeth, with the anchor covering $4\frac{1}{2}$ spaces between the tangents.

Lift 2°; minimum amplitude of the lifting arc of the pendulum, 3°.

The radius of the wheel is taken as a unit of measurement.

Radius of the wheel = 1·0000	A E = 0·2404
Distance between centres = 1·1223	B C = 0·4133
A B = 0·6604	O C = O C′ = 0·5095
A O = 0·5509	O E = O H = 0·6054
B O = 0·6801	O D = 0·4137

Recoils.

Disengaging { A F = 0·2832, O F = 0·3066, F C′ = 0·2096 } Engaging { B H = 0·6459, O H = 0·6054 }

The engaging recoil face, strictly speaking, should be concave, having a radius of curvature equal to 5·2454. It will be evident that we err very little from the truth if we make it straight.

Measurements for verification.

C C′ = 0·9120	D C′ = 0·8178
D E = 0·9038	D C = 0·0981
C E = 0·9880	C′ E = 0·0987

These last figures are very useful for testing the accuracy of the several dimensions, either during the progress of the work or after completing and hardening the anchor.

958.—*To draw the calliper of the anchor.*—On a specially prepared plate draw the line A B (fig. 7). Then point and drill holes at A and B, their distance apart being given by the table.

From these points with the radii A O, B O, determine the centres O and O′ of the two anchors, and drill holes at these points.

From the same centres A and B, and with radii A E, B D, describe the circumferences that determine the impulse faces of the pallets.

Now from the centre O′ describe, with the three radii O D, O C, O E, the arcs that fix the thickness of the pallet-arms in the dead-beat escapement, and its calliper will then be complete.

To form a recoil escapement, it will be necessary to:

Describe two intersecting arcs from the points A and O as centres, with radii A F, O F, and from F, where they cross, with the radius F C′ draw the circumference that determines the disengaging recoil face.

Then with B as a centre, at a distance B H describe a circular arc; its intersection with another arc drawn from the point O, at a distance O H (equal to O E), gives the point H, which is joined by a straight line with C.

959.—It will be seen that nearly the entire calliper is formed of circular arcs; it is therefore of very great importance that they be drawn accurately, and this can be done without much difficulty if the following precautions are taken. After approximately setting the opening of the compass to the required amount, draw on a spare piece of metal, with a hole equal to that in the calliper for a centre, two arcs opposite to

each other, and then ascertain, by measuring the space between them, whether it is exactly double of the required radius, as given in the table.

960.—A simple inspection of the figure will show that this system of anchors can, with the help of M. Rozé's tables, be made by machinery with very great accuracy. Thanks to the economy of force effected by using concave impulse faces, accurately formed, M. Rozé has been enabled, by employing springs that admitted of a greater number of turns in the winding up, to make ordinary timepiece movements go for a month that are usually only expected to go for sixteen or eighteen days.

961.—*Principal dimensions of a Rozé escapement*—that maintained a uniform rate for a long period.

Diameter of movement, 4 inches (with centre seconds); going for one month.

Diameter of escape-wheel, 22 millimetres (0·866 ins.),—30 teeth.

Anchor covered $4\frac{1}{2}$ spaces.—Lift, 2°.—Mean supplementary arc, 2° 30′.

Total arc, 4° when the spring is nearly run down, becoming 6° degrees when wound up. (The suspension rendered the vibrations isochronal.)

Pendulum: weight, 940 grammes (33 ounces).—Virtual length, 248 mm. (9·8 inches).

Ratio of length of escapement arm:

To weight of pendulum ... ... : : 1 : 170
(comparing millimetres and grammes).
And to virtual length ... ... : : 1 : 45

Half Dead-beat anchor escapement.

962.—What is known as the half dead-beat or the half-recoil anchor, from the fact that recoil only occurs against one of the pallet arms, the wheel resting on the other arm against a locking face concentric with the pallet axis, is similar to the anchor shown in figure 5 (plate XII.), except that the excentric curve *g h* is replaced by a circular arc *g m*.

Figure 9 of the same plate represents an escapement of this class.

963.—*To draw the calliper of a half dead-beat anchor.*—The calliper is drawn as in the case of a recoil escapement. (**941-2**), the only differences being, as will be seen by an examination of this figure, that: (1) the engaging pallet, as already

stated, has a circular locking face *c d*, against which the tooth rests without recoil; (2) the wheel is inverted, that is to say acts with the straight faces of the teeth.

Recoil on a single face has an effect equivalent to half the amount of recoil on each of the two pallets. As the total recoil must only be of slight amount, there is some difficulty, owing to the methods of construction usually adopted, in preventing its being either excessive or zero on one at least of the two arms when both occasion recoil; it therefore becomes much easier to adjust when only caused by a single straight plane.

Two points require to be exactly proportioned, the weight of the pendulum bob and the angle of lift; for the degree of inclination of the recoil face, *a b*, is dependent on this lift (which usually measures 5° or 6°). A few trials made with care will enable a manufacturer, who is thoroughly cognizant with the laws of our new theory, to determine the most suitable proportions (**917—935**).

964.—Very good results for ordinary purposes have been secured with this escapement, and it was much used at the time when silk suspension was in favour, since it requires the suspension to be very pliable; but the one is gradually dying out with the other.

The causes that have brought about this disuse, in addition to those already indicated, are: The escapement can only with great difficulty be made of large dimensions so as to be used with a spring suspension and heavy pendulum; it needs rather more care in construction as well as greater exactness in its several proportions.

965.—*Detailed proportions of a half dead-beat escapement.*—(Rate very satisfactory for ordinary purposes.—Clock made by M. H. Robert.)

Diameter of movement, 80 mm. (3·2 ins.).

,, of the wheel, 18·5 mm. (0·73 in.),—37 teeth.

Anchor covered 4½ teeth; the circle *a b c* includes the points of 5 teeth; lift 6°.—Supplementary arc on either side, 1°·5.

Total arc, 9°;—varies slightly as the mainspring runs down.

Pendulum.—Weight, 17 grammes (just over ½ an ounce). Virtual length, 164 mm. (6·46 ins.).

The ratio of the length of pallet arm is to:

Weight of pendulum about ... ... : : 1 : 4·5

(comparing millimetres and grammes)

and to the virtual length of the pendulum : : 1 : 42.

ANCHOR ESCAPEMENTS WITH SLIGHT RECOIL.

Roller anchor or Brocot escapement.

966.—The plan of this escapement, invented by MM. Brocot, is shown in figure 61.

The setting or anchor proper is made of brass, and two rollers project from it, being let in at E A and R B. They are cut away to half their thickness. The locking takes place when a tooth rests against the highest point of the semi-circular face of the roller, for instance at N, and the impulse is occasioned by the pressure exerted by the extremity of the tooth against the inner portion, N E, of the curved face of the roller, when in motion from the axis of the wheel.

As was observed by M. Redier, who published a discussion of this escapement in the *Revue Chronométrique*, those who are anxious to criticise complain that in it the lockings are untangential and the recoil and lift irregular.

There would be some point in these objections if the escapement were proposed for use in astronomical clocks; but no one, so far as we are aware, ever suggested that it should serve so dignified a purpose.

Although it has one really weak point, namely a loss of force occasioned by the accelerated movement of the wheel towards the conclusion of the lift, this escapement has compensating advantages in presenting no difficulties of construction; however little care is devoted to its construction, the oil is retained on the acting surfaces; it is very easy to guarantee its lasting by making the rollers of some hard stone; and, finally, its extensive adoption for a good many years past proves that, if made with intelligence, it will give results that are perfectly satisfactory for ordinary purposes.

967.—*To draw the calliper of the Brocot escapement.*—Assume the number of teeth of the wheel, as well as the number covered by the anchor to be known; let, for example, the wheel have 30 teeth, and the anchor include $10\frac{1}{2}$ teeth between the two locking faces, or 11 if we count from the beginning of one lift to the conclusion of that which succeeds.

Draw the circumference of the wheel. Mark on it the positions of three teeth, namely: a first, x; a second, the next to its right, N; and, lastly, a third tooth, R, the 11th to the left, counting from N inclusive.

Draw the radius O R through the point R, and another

radius O E through the middle of x N. The intersection of this latter with the circumference gives the position of the centre of the roller or pallet. Mark this point and describe the half-circle of the roller, taking as a radius half the interval between the points of two successive teeth.

Now mark the point E on the straight face of the half roller. The distance between this point and the edge is rather less than half the radius, or something under one quarter of the

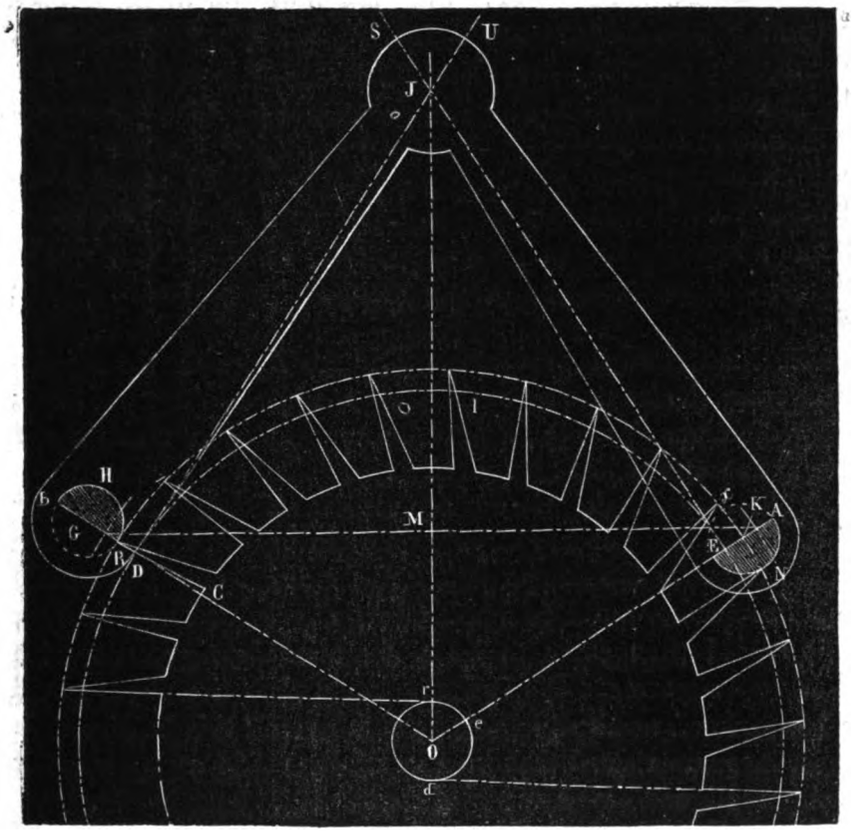

Fig. 61.

maximum thickness of the roller; with the radius O E describe a second circle D I E. The portion of the roller that overlaps this new circle gives a measure of the displacement of the pendulum during the supplementary arc on one side.

At the points D, E, where the circumference D I E intersects the two radii O D, O E, erect the perpendiculars D U, E S; the point J where they meet is the centre of movement of the anchor.

The second roller still remains to be drawn; but this is easy,

since it is only necessary to take a radius equal to half the distance between two teeth and describe a semicircle with a centre on O R prolonged, such that the roller B H R is tangential to the circumference of the wheel.

All the proportions being now known, the anchor can be roughed out; some adopt the form indicated in the figure, while others make it like an ordinary Graham anchor. (The concluding observation of article **999** is applicable to these rollers.)

968.—It will be noticed that the tangents which fix the centre of the anchor are not drawn from the points of teeth, because the part of the tooth that rests against the roller is variable, gradually moving from the extremity towards the centre of the wheel through a distance measured by the supplementary arc; the resting point of the tooth thus coincides successively with a series of concentric circles. We have selected the circle D I E as a mean value, and it corresponds to half the path of the anchor while performing the supplementary arc, counting both oscillations.

If it be desired to prevent sensible recoil during the supplementary arc, the front faces of the teeth must be slightly inclined; their projections *d*, *r*, etc., should be tangential to a circle *d r* having the same radius as the roller.

969.—By forming the curved surfaces of the rollers against which the locking occurs of certain definite forms, it would be possible to make the escapement strictly dead-beat; but we do not believe there would be much advantage gained by doing so, for it would then come to be a dead-beat escapement with relatively long arms. We feel convinced that the slight recoil can be made of service in timing; and, even if this be disputed, it must be remembered that in a carefully made escapement it is quite negligeable as a cause of wear.

Even when the front face is directed towards the centre of the wheel the recoil will only be, approximately:

For 1°	of supplementary arc	on	either	side, 0°·2
,, 2°	,,	,,	,,	0°·5
,, 3°	,,	,,	,,	0°·9.

970.—*Details of a Brocot Escapement*,—going well.

Diameter of movement, 105 mm. (4·13 ins.).

Diameter of escape-wheel, 20 mm. (0·79 in.),—30 teeth.

Anchor covered $10\frac{1}{2}$ teeth between the tangents.

Lift, 3°. Supplementary arc on each side, about 2°.

Total arc, 7°,—varying as the spring runs down.

Pendulum: weight, 500 grms. (17·6 oz.). Virtual length, 248 mm. (9·76 ins.).

Length of escapement arm, about 17·5 mm. (0·69 in.), is to:

The weight of pendulum ... ... :: 1 : 28
(comparing millimetres and grammes)
and to the virtual length of pendulum :: 1 : 14.

Recoil Brocot escapement with two wheels.

971.—This escapement (fig. 2, plate XII.) is so arranged that it can be made dead-beat or have varying degrees of recoil. It appears to have been made as a practical solution of the problem of introducing recoil into dead-beat escapements.

The two escape-wheels, having the same diameter and number of teeth, are mounted on the axes of two identical wheels driven by an ordinary train.

There is only one pallet, and it is so mounted on the pallet axis that it can be raised or lowered without its centre of movement being thereby altered.

If now the acting faces of the teeth have a curvature concentric with the centre of movement of this pallet, the escapement will be dead-beat, and the recoil will become more and more pronounced as the pallet is made shorter.

The figure indicates the relative dimensions that were found to secure isochronism of the oscillations of the pendulum in a clock exhibited at the Exposition of 1839.

The following results were obtained with it:

When made dead-beat the escapement gave a difference in the rate of 17 seconds in 24 hours with the motive force at one time reduced by one half and at another time doubled; when set to give a certain amount of recoil (indicated by previous experience), with the same variations in the force applied, the change in the rate was reduced to 1 or 1·5 seconds.

The recoil escapement always gave longer arcs of oscillation than the dead-beat.

Robert's anchor escapement.

972.—Anxious to facilitate the manufacture of good ordinary clocks, M. H. Robert adopted the form of anchor escapement shown in fig. 12, plate XII. He only used one size

of escape-wheel, but adapted to it three forms of anchor, covering 5, 6 or 7 teeth according to the size of movement and pendulum employed. These forms are described with full details in Vol. I. of the *Revue Chronométrique*.

The anchor is made of brass, the stone pallets being set in it.

The object he had in view will be at once evident: The wheel being of a constant size, the anchor made of brass and the stones with straight faces, can all, with the aid of suitable machinery, be made mechanically in large quantities and as accurately as desired.

973.—It might perhaps be objected to this escapement that the substitution of a straight for a curved face gives rise to a slight recoil. Our readers will remember that, so far as concerns ordinary timepieces, we do not consider such a recoil of any moment; but, notwithstanding this, M. Robert thought it advisable to reply to such a criticism, and he pointed out that, with a very limited lift (1° or 1°·5), straight and curved locking faces would so nearly coincide that there would be no appreciable effect of recoil, and even with a lift of several degrees the movement would still be hardly perceptible.

The following table, moreover, calculated by M. Robert, gives the value of the recoil, the radius of the wheel being assumed 100 mm.

Lifting arc.	Secant.
5°	100·382
4°	100·244
3°	100·137
2°	100·060
1°	100·015
0°·45	100·008
0°·30	100·004
0°·15	100·001.

Since the recoil is measured by the difference between the radius and secant, we have:

For 5°, $\frac{382}{1000}$ or over $\frac{1}{3}$ millimetre (0·015 in.).

For 1°, $\frac{15}{1000}$ mm. (0·0006 in.).

But we have assumed the radius of the wheel 100 mm., whereas it is ordinarily about one-tenth this amount; hence the above figures must be diminished by about nine-tenths.

DEAD-BEAT ANCHOR ESCAPEMENTS.

Ordinary form.

974.—Graham* changed the shape of the anchor from that adopted by Clement and made it cover a greater number of teeth; he replaced the excentric curved faces, producing recoil, by surfaces concentric with the axis of rotation.

As in the case of watch escapements, the impulse plane is at times entirely on the anchor or on the wheel and at other times in part on each.

The latter arrangement is mainly adopted with a view to make the tooth more solid and to ensure the retention of oil. As regards its action, the escapement may be assumed to be identical in the two cases; whether the plane is entirely on one or the other part or divided between the two.

975.—The small dead-beat anchor is made in the same manner as the half dead-beat anchor (fig. 9, plate XII.) except that the left-hand pallet is cut away along the arc *s o*; thus the recoil face is replaced by a concentric locking face.

976.—In an escapement that has the impulse plane divided between the teeth and anchor (fig. 13 plate XII.), the latter will have precisely the same form as that just considered; the only difference will consist in the thickness of the arms being reduced by an amount corresponding to the portion of the impulse plane that is transferred to the teeth. Its calliper must be drawn in the same manner as that of an escapement with concentric locking faces, except that the several angles have different values; the lifting angle of each arm will, of course, consist of two angles *a* and *v* corresponding to the two inclined planes. The real lift will therefore differ from the apparent lift.

977.— As has been already shown (**900**), we cannot expect to secure good results with a dead-beat escapement unless two conditions are satisfied: its several dimensions must be in strict proportion to the pendulum, and the whole must be accurately made. This circumstance will suffice to account for its want of success in ordinary timepieces, at least when constructed on the usual system by average escapement-makers; but it also shows

* George Graham was the most celebrated of English horologists. He was born in 1675 and died in 1751, and invented the escapement that bears his name, the cylinder escapement for watches, the mercurial compensation pendulum, besides being the first to suggest the gridiron pendulum. He was both learned and skilful, and his reputation was still further enhanced by the improvements he introduced in certain astronomical instruments.

that the dead-beat escapement may render good service if made with appropriate machinery under the immediate eye of a skilled maker.

Anchor Escapement for Regulators.

Graham's Escapement.

978.—This form is most frequently employed with an escape-wheel of 30 teeth carrying the seconds hand at the extremity of its axis, which thus gives 60 beats per minute; or the wheel may have 60 teeth and the pendulum beat half-seconds.

Our Theory explains the manner in which the length of the escapement arms and the size of wheel may be determined when the dimensions of the pendulum are known (**1018**); we shall, then, merely add that, according to an empirical rule, the number of teeth covered by the anchor is (in the callipers adopted at the present day) between a quarter and a third the circumference of the wheel (**1019**). If this amount is exceeded the thickening of oil has an appreciable influence, and, with a less amount, very accurate workmanship is necessary; it being extremely difficult to make sure that differences in the force acting on the very short locking faces are avoided.

979.—In a word, it is necessary to remember that uniformity depends on a number of terms, and we cannot repeat this fact too often; no one alone can ensure success, for it is under the combined influence of all the others. Among these several terms, moreover, some remain constant, such as the lifting angle, the length and weight of the pendulum, etc., whereas others alter with time; for example, the resistance of oil, the supplementary arc, the action of suspension spring; an action which varies with the extent of arc of oscillation, etc., according as the spring is more or less adapted to secure isochronism.

As evidence that success depends on the judicious combination of the entire escapement and not on any single condition, however important, it will be sufficient to point out that, with a given suspension, the influence of flexure differs according as the escapement arms are long or short, and that the progressive change in the influence of oil will depend on the extent of the supplementary arc.

980.—The question whether the locking faces or the middle points of the impulse planes should be equi-distant from the centre of movement, has given rise to discussion as regards this escapement also, and we cannot do better than refer the reader to articles **687** to **693**, from which he should select the passages

bearing on the subject; we would add that many skilful makers adopt the two forms indifferently, and that, in the following drawing, the locking faces coincide with the two tangential points.

981.—*To draw Graham's anchor escapement.*—The design is similar to that of any other dead-beat escapement, the only difference consisting in the choice of lifting angle and of the number of teeth covered by the anchor. Having regard to its importance we will again give the details of the drawing.

Knowing the diameter of the wheel, which is assumed to have 30 teeth, draw the circumference *c r a* (fig. 8, plate XII.). Mark on it the point of one tooth *a* and of the eleventh to the left *c* (assuming ten teeth to be covered by the anchor when the wheel rests against the internal locking face).

Draw the radii G *a*, G *o*, the latter passing through the point midway between the extremities of the tenth and eleventh teeth, and at the points *a*, *o*, erect perpendiculars *a* B, *o* B, meeting at B, the centre of movement of the anchor. From this centre with a radius B *a* describe the arc *h t d*, which determines the two locking faces; it will pass midway between the points of the tenth and eleventh teeth. Then from the same centre B draw two other arcs to fix the thickness of the pallets; one, *g c*, passing through the eleventh tooth, and the other, *n s*, bisecting a line that joins the points of the teeth *a*, *n*.

It only remains to mark off the lifting angle, which is, as a rule, 1° in astronomical or regulator clocks, and 1°·5 to 2° for half-second regulators.

Draw the lines B *n m*, B *d p*, forming the lifting angle with the tangents; join their points of intersection with the circular arcs by straight lines; in other words, join *a n* and *c d*.

Draw *a f*, forming with the radius *a* G an angle equal to the inclination of the front face of the teeth. This angle should be sufficient to avoid all risk of adhesion, etc.; it varies from 6° to 12°. The lesser of these is adopted for small escape-wheels or such as have many teeth, in order to ensure sufficient solidity at their base. As to the straight portion of the back of the teeth, its inclination is determined by the circular arc *n s*, and is such that the corner *n* of the anchor cannot come in contact with the back of a tooth; the same may be said of the concave part of this face, for its form is determined by the inward motion of the corner *c* during the lift on *a n*.

The remaining details will be easily filled in from what

follows, an inspection of fig. 8, and such portions of article **1001** as are applicable here.

982.—Some makers construct the anchor entirely of one piece of steel, others, more numerous, make a brass setting with steel pallets fixed by screws and steady-pins; they add the screw *z*, and sometimes a second opposing screw, by the aid of which it is possible to slightly modify the opening of the pallets should it be necessary. Whenever the escapement is intended for an astronomical clock, the pallets are fitted with hard stone, usually oriental sapphire, for it has been observed that, in time, steel anchors always wear at the contacts.

The acting surfaces of the anchor are rounded crosswise or "beaded" (**1002**).

An anchor entirely of steel will be distorted in hardening unless only the pallets are hardened, and even then it is essential to leave a little metal in excess at the parts most likely to be distorted during the operation, to enable the workman to correct by smoothing, after trying the escapement. Or the anchor may be closed a little, if needful, after letting it down to a blue colour; but, when it is not intended to supply the pallets with rubies, they should be held by screws between two small metallic blocks during the heating, so as to prevent their changing colour and thus losing any of their hardness.

983.—The teeth of the wheel (viewed sideways) are formed indifferently as shown in fig. 8, plate XII., or as at *a* fig. 53, page 493.

The section of the tooth, looking at its front face, is nearly always of the form shown at A in the same figure 53. Some makers drill a small hole in each tooth to facilitate the retention of oil, a method adopted by the elder Breguet, L. Berthoud and Motel.

Pointed teeth of brass were found to wear when subjected to the severe pressure that occurs in turret clocks; with a view to avoid this inconvenience, a portion of the impulse plane has been transferred to the teeth, which thus have the form shown in fig. 14, plate XII. Attempts have also been made in heavy clocks to transfer the incline entirely to the teeth; but in that case the extremity of the pallet was observed to wear.

Curved and Straight Impulse faces of the anchor.

984.—By forming the lifting faces to correspond with certain curves in the direction of their length, convex or concave

according to circumstances (in other words, curves so formed as to satisfy the double condition of securing a certain progression in the velocity of the wheel and a reduction of the impact on the locking face), it is possible to slightly diminish the amount of motive force absorbed; but these conditions give rise to two difficulties, the discovery of a mechanical means for accurately forming the curves, and the prevention of any damage to the points of the teeth when the pressure of the pallets acts directly against them, the wheel being uninfluenced by the motive force while the pendulum continues in motion. In the best class of clocks the pendulum rod must not be held with friction in the crutch.

Moreover, this reduction in the shorter drop renders the blow that marks the second very indistinct, a fact which is objected to by some observers.

What has done most, however, to hinder the success of these curves is the fact that the majority of those experimented upon have been drawn in accordance with that erroneous and obsolete condition of *equal lifts;* that is to say, successive equal portions of the motion of the two mobiles correspond (**672**).

985.—*Details of a regulator with anchor escapement.*—This escapement is of the form shown in fig. 8, plate XII., the regulator being of M. Breguet's excellent make.

Diameter of wheel, 40 mm. (1·58 ins.);—30 teeth.

Anchor covered $10\frac{1}{2}$ teeth between the two tangential points.

Lift, 1°;—supplementary arc on either side, 30′; that is to say, a total arc of 2°.

Escapement arm, 38 mm. (1·5 ins.).

Pendulum; weight, 8·6 kilogr. (18·9 lbs.),—beating seconds.

Proportion of length of escapement arm to

Weight of pendulum ... ... : : 1 : 226

(comparing millimetres and grammes)

and to virtual length of pendulum : : 1 : 26

The motive power was a weight of 3·6 kilogrammes (7·9 lbs.).

Various modifications of the anchor escapement.

986.—Perron* attempted to modify the anchor escapement of regulators by placing the impulse plane entirely on the teeth

* An expert horologist, born at Besançon in 1779. He published an *Essai sur l'histoire abrégée de l'horlogerie* which shows him to have been specially impressed with the writings of F. Berthoud.

of the wheel and substituting rollers, free to rotate on pivots, for pointed pallets (fig. 5, plate XI.). As it was impossible to make the rollers and pivots of extreme fineness, the drop was necessarily augmented by half the thickness of the roller; moreover, although the surface of the roller might have done without oil, such could not have been the case with the pivots; thus, as has been observed by a recent writer, the difficulty was merely transferred to another locality.

A provincial clockmaker has also suggested the arrangement shown in figure 62 as an improvement on the ordinary Graham escapement.

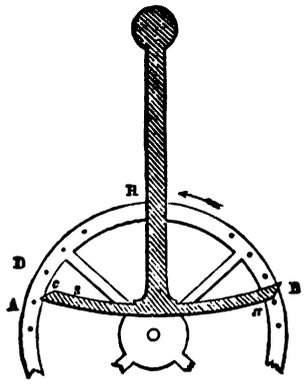

Fig. 62.

This combination of the anchor and pin escapements is by no means a happy one; it retains all the objections of Graham's form while wanting the advantages of the pin escapement. The piece H is just as difficult to make as the ordinary anchor, for it is not on the mere cutting out of a particular form that most time is expended; the wheel necessitates more labour than does an ordinary flat wheel divided with a cutter, the pins are ill adapted to retain oil, and, unless half the thickness of each pin is cut away, the drop will be needlessly increased. Moreover the form of the pallets is such as to increase the tendency of the oil to leave the pins and pass towards the vertical arm H; it will inevitably thus pass away if supplied in any excess.

987.—An arrangement (fig. 3, plate XI.) rather similar to the preceding as regards the form of the pallets, but differing in that the crutch was entirely suppressed and the anchor inverted (thus retaining oil on the acting surfaces better), was attempted as early as 1824 by a clockmaker then quite young, M. Vérité. He only employed it in the best class of clocks; for he ascer-

tained that most perfect workmanship is essential, since any alteration in the distance between the centre of the wheel and the centre of suspension of the pendulum might cause the escapement to catch.

This modification, which led its inventor to definitely suppress the crutch in his clocks, was suggested to him by the cardboard clocks, made about 1820 by M. Duclos. Fig. 12, plate XI., represents one of these escapements. The pendulum carried an inverted anchor at its upper end, the pallets of which were faced with horn and acted without oil against the teeth of a wheel made of some specially prepared cardboard. The method of making such clocks is now lost, although, greatly to the astonishment of clockmakers, many of these singular machines have gone without sensible change and with a striking approximation to exact time.

Kessels' anchor escapement.

988.—Kessels,* of all the more celebrated makers, was the one who employed the shortest escapement arms. His anchor covered six teeth, measuring between the two locking faces, or seven counting from the commencement of one impulse to the termination of the next succeeding, the wheel having 30 teeth. A portion of the impulse face was on the teeth.

Figure 9 of plate XI. clearly shows his arrangement; it is drawn in precisely the same manner as the Graham escapement above described, except that the middle points of the impulse faces are on the tangents that determine the centre of the anchor, and, in making the drawing, it is necessary to allow for the diminution of the drop owing to the greater thickness of the extremities of the teeth; these are slightly inclined backwards, the front corners being formed into a sharp solid angle.

989.—Several clockmakers have made regulator escapements of the form recommended by Kessels; but they have not been equally successful, and the explanation of this fact is not far to seek. Kessels carried the reduction of the arms, and consequent increase of pressure, to the farthest practicable limit; so that, for equal success, it is absolutely essential that the minutiæ in the workmanship be identically the same as his. Any slight difference in the play of pivots, friction, impelling force or velocity of the wheel, acting lengths of the locking face, nature

* A clever German clockmaker who died in 1849. He worked for a long time with Breguet and afterwards settled at Altona; he there made a great number of excellent astronomical clocks for German, Swiss and Russian observers.

of suspension spring, etc., would thus acquire an importance that is relatively greater than it would be were the anchor of rather greater dimensions.

990.—*Details of a Kessels escapement.*—

Diameter of escape-wheel, 40 mm. (1·58 ins.),—with 30 teeth.

Anchor covered 6 teeth between the locking faces.

Lift 1°·33 (1°20′); supplementary arc, 20′ on either side; that is, the total arc is 2°.

Escapement arm about 12·6 mm. (0·5 ins.). The pendulum beat seconds and weighed about 5 kilogr. (11 lbs.).

Proportion of length of escapement arm:

To weight of pendulum about ... : : 1 : 396
(comparing millimetres and grammes)
and to virtual length of pendulum : : 1 : 78.

Vulliamy's anchor escapement.

991.—Figure 2 of plate XI. represents this form of escapement, which only differs from the preceding in the form of its escape-wheel teeth, and in being provided with two micrometer screws, *c*, joined together and of slightly different pitch; by its means infinitesimal changes can be made in the distance between the arms, and when brought to the exact positions they are clamped by the screws *s* and *t*; the two arms move in contact with each other with stiff friction, the screw *c* passing through two parallel spurs.

The anchor covers 8 teeth when a 30-tooth wheel is employed, measuring between the two tangents, which are made to pass through the middle points of the impulse faces.

The lifting angle is 1°.—For a wheel having a diameter of 40 mm. (1·58 ins.) and a seconds pendulum, the lever arm is to the virtual length of pendulum as 1 : 50, and as 1 : 33 with a wheel 62 mm. (2·44 ins.) in diameter (as shown in a drawing given by Moinet).

Winnerl's anchor escapement.

992.—Moinet observes that: "F. Berthoud suppressed the crutch in a clock beating half seconds, which he suggested but never carried into execution. In other cases the crutch has been suppressed of necessity but not with satisfactory results; thus this change, which possesses real advantages, ill understood by these makers, was not again attempted."

More recently M. Winnerl, in his efforts to improve the better class of ordinary clocks, has made escapements without

the crutch. Fig. 1, plate XI. represents one of the two forms he has employed; the second, which is not shown, is only used in certain special cases; it has the escapement arms inverted and set above the point of suspension.

In both the anchor is fixed to a block below the suspension spring; this, then, amounts to fixing it to the pendulum rod itself. Several changes result from the suppression of the crutch and its pivots: there is less weight to be moved; the influence of the oil on the crutch pivots vanishes, as well as the shake occasioned by the pendulum striking the crutch fork; and, lastly, the centres of movement of the pendulum and anchor are identical. The diminution in the number of moving parts and the friction has a sensible effect, for, with an equal lifting arc, the oscillations are rather more extended.

It is essential when this system is adopted that there be very accurate workmanship and that no more oil be applied to the pallets of the inverted anchor than can be retained with certainty by adhesion; this condition is absolutely essential with a view to retain the oil at these acting surfaces.

The drawing of this escapement can be made as in other cases, the centre of the pallets coinciding with the centre of flexure of the suspension spring. When the proportions are as shown in fig 1, plate XI., the centre of movement is at the middle of the length of the spring; if this spring is weaker, it will be somewhat higher, at about one third the length from the top. The centre does not, however, remain the same with pendulums of different weights; at the same time it is never lower than the middle point of the spring and this may be taken as the most usual position. The anchor, when cut out, is screwed to the lower block of the suspension spring, taking care that the small hole drilled at its centre coincides with the line of flexure of the spring; and, when all the parts occupy the positions indicated in fig. 1, the suspension of the pendulum is fixed.

Observation.—M. Winnerl finds that if these pendulums are set vertically in the cases, nothing is required to set the escapement in beat; but, if any means of adjustment should be desired, such an arrangement as is shown in fig. 4, plate XI., may be adapted to the pendulum.

On the prolongation *d e* of the suspension hook the pendulum-rod *f* is fixed by a screw; this rod is provided with two projecting pieces at *g* between which passes the prolongation

e of the hook. A screw with a milled head rotates between these two projections as bearings, engaging in the piece *e*, which can thus be caused to move to the right or left.

Identical escapement by M. H. Robert.

993.—This escapement, intended by its inventor principally for use with half-second pendulums, is shown in fig. 10, plate XI. It is said to be *identical* from the fact that both the anchors and the wheels can be made absolutely alike, that is interchangeable.

A steel ring *b d* (shown both in front and side elevation) is cut away so as to allow the wheel to pass. It forms an anchor with concentric lockings. The apparent total lift (**307**) is 6°, or 3° on either side of the vertical. The lockings are tangential.

This ring is turned both inside and outside, and then fixed by friction on the support *f*, which forms part of the axis of the anchor.

The wheel is flat and of hardened steel, perfectly polished at the points that lock and on the impulse faces.

The two locking arms may be assumed to be equal, for they only differ by about the $\frac{1}{50}$th part.

The hardened steel wheel is very firm and retains the oil well; but its construction, as well as that of the ring, and the various adjustments, necessitate very great care if accurate workmanship is aimed at.

The end M. Robert had in view was to design an escapement beating half seconds that had the escapement arms sufficiently short and at the same time gave a perfect locking action; results which can only be ensured by employing the lathe.

994.—*Details of an identical escapement* that has gone for 30 years; (as a rule these escapements give a variation in the rate of less than a minute in fifteen days).

Diameter of the movement, 90 mm. (3·54 ins.). Spring suspension.

Diameter of wheel, 21 mm. (0·83 ins.),—30 teeth.

Anchor covers 6 teeth.—Lift, 3° on either side of the vertical. —Supplementary arc, 2° on each side;—total arc, 10°.

Pendulum; weight, 250 grms. (8·8 oz.). Virtual length, 248 mm. (9·76 ins.).

The ratio of the length of escapement arm (7 mm. or 0·28 ins.) is:

To the weight of pendulum ... : : 1 : 35·7.
(comparing millimetres and grammes),
and to the virtual length ... : : 1 : 35·4.

Pin Escapement.

For regulators and turret clocks.

995.—This escapement is no more than an anchor escapement with the two arms brought near together; but such a closing up, when the anchor and wheel move in parallel and not in perpendicular planes, can only be accomplished if the wheel have pins planted in its face instead of teeth cut in the flat: it is for this reason that it received and has retained the name of pin escapement, a name that is by no means distinctive, since there are several other forms that involve the use of a wheel of this description.

The arrangement usually adopted at the present day is shown in fig. 6, plate XII. Its mode of action will be evident: when the pendulum oscillates towards the right, the pin *a*, resting on the locking face *b*, arrives at the entrance edge of the incline *o c*, and, in traversing it, gives an impulse that is followed by the locking of the pin *d* against the face *i*. On the return of the pendulum, this pin *d* leaves the locking face *i* and passes over the impulse face *i a*, when it falls on to the face *b* and is locked; and so on.

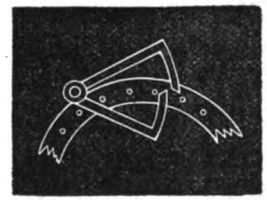

Fig. 63.

The invention of the first pin escapement was published in 1741 by Amant, a French clockmaker. Fig. 63 represents this very remarkable discovery; remarkable because the pin escapement as employed at the present day differs in no way from that of Amant, except in having the arms somewhat longer and in the substitution of semi-cylindrical for cylindrical pins.

996.—Lepaute* took up Amant's escapement and modified it. With a view to diminish the drop he made the change just indicated in the form of the pins; he made the arms pass

* J. A. Lepaute was born at Montmédi in 1709 and died in 1789; he was one of the most celebrated of French horologists. He did much to improve his art, especially in regard to turret clocks. We are indebted to him for a *Traité d'horlogerie* which was formerly much thought of. His wife, a very clever mathematician, helped him in some of his labours. Jean-Baptiste, his brother and pupil, as well as his nephews, have honourably maintained the reputation he acquired.

one before and one behind the wheel (fig. 1, plate XII.) and employed a double row of pins, so that the two lockings as well as the two impulses occur at equal distances from the centre of movement of the anchor; this precaution was, however, not of so much importance as Lepaute considered it, for the difference in the action of the two arms, when considered in relation to the mass moved, is very slight, and its influence on the rate is inappreciable. The modern form, shown in fig. 6, plate XII., gives as good a rate as can be obtained with Lepaute's arrangement, and in addition offers greater facilities of manufacture and adjustment; this will be at once evident from a comparison of figures 1 and 6.

In Lepaute's description of his escapement we read:

"By lengthening the arms *we secure an increased force for the impulse*, but by reducing this length we bring the locking faces nearer to the centre of movement and therefore diminish the friction." This second statement is true but the first is erroneous (**219**), and this mechanical error has led the manufacturers of the Jura to provide their clocks with escapements, the arms of which are absurdly long. Lepaute recommended one and a half times the diameter of the wheel as the best length, but even this is, as a general rule, too great.

997.—M. J. Wagner was, we believe, the first to point out the advantages that would be secured by reducing the length of the arms with this form of escapement.

We have indicated the conditions that determine the length of the escapement arms (**978**); we would only add, as a first approximation or empirical rule, that, in the callipers actually made, this length is usually from one to one and a half times the radius of the circle that passes through the centres of the pins; but with regard to this estimate, which only applies to the calliper ordinarily met with, the reader must be careful to bear in mind that an arm which is too long occasions a loss of force and irregularities due to changes in the condition of oil, while a short arm renders very accurate workmanship essential and is liable to give rise to more rapid wear of the acting surfaces, owing to the increased pressure (**918**, **979**).

998.—*To draw the calliper of a pin escapement.*—The wheel is assumed to have 30 teeth or pins if the pendulum is to beat seconds (or 60 for a half-seconds pendulum, but in this case the wheel is large or the impulse planes very short).

Draw the line H *a* (fig. 6, plate XII.), and from H as a centre, with a radius equal to that of a circle passing through the middle points of the pins, describe this circumference. Mark the middle point *a* of the first pin; then the centres of several others above and below. The angular distance apart of these points is 360° divided by the number of pins in the wheel; thus for a 30 pin wheel, the angle will be 12°, which may be traced with the wheel-cutting engine, if one is accessible; if not, with a protractor.

Draw the semi-cylindrical pins, whose centres have thus been determined; their radius will be one quarter the distance between two consecutive centres.

Through the point *a* draw the perpendicular L N, tangential to the circle of pins; mark on it the centre of the anchor, say at N. From this centre and with the radius N *a* describe the arc *s*; then with the same radius increased by that of a pin, draw the arc *b*. Now draw in the two arcs *i* and *t* at distances from the first two exactly one quarter the interval between two centres. If the calliper has been drawn with care, the semi-cylindrical pin will just pass without play between the arcs *s* and *b*, and, if a pin were resting against the locking face *i*, the pin immediately below it should pass without play against the arc *t*, in the manner indicated in fig. 15. In making the escapement, the thickness of the pins must be slightly reduced on their flat face in order to ensure the requisite freedom, if the smoothing of the upper and under faces of the two pallets is not sufficient.

999.—Having previously decided upon the total lifting angle (usually 2°; 1° for the incline and 1° for the half pin, **1010**), draw the two angles L N J, J N K, on the same side of the vertical L N, together equal to the lifting angle, the first being the amount due to the rounding of the pin, and the second to the incline of the pallet. Join the point *o* at which the line L N cuts the half-circle of the pin *a*, with the point *c* where the line J N cuts the arc *t*; we thus obtain the impulse plane *c o*.

Join the left corner of the pin with the point of intersection of the line K N and the arc *i*, and we have the second impulse plane *i a*. All the principal dimensions of the escapement are now determined.

Remark.—Since the interval between the centres of two consecutive pins is equal to twice the thickness of a half pin plus twice that of a pallet, it follows that we can, if necessary, give, say, a greater thickness to the pallets and a less thickness

to the half pin; thus this latter might have only one third or one quarter the thickness of a round pin, but it should be struck from a centre higher than the point *a*; for the curved face of the pin is one of the elements that determine the lifting angle, and this angle would therefore be modified by diminishing or increasing the size of the pin in the direction *s b*.

1000.—The lift in this escapement can thus be regarded as consisting of two parts; the first is produced by the action of the small curved face, *o b*, of the pin against the engaging edge of the impulse face, and the second by the corner, *b*, of the pin passing over the plane.

When these two parts are unequal it is necessary to draw a distinction between the real and the apparent lift. They differ when the right and left-hand lifting angles overlap instead of following immediately one after the other (fig. 13, plate XI.).

Observations and practical details.

1001.—After drawing the inclined faces of the pallets with the greatest possible accuracy, prolong them to *g* and *f* (fig. 6, plate XII.), and from the centre N describe two circular arcs to which these lines are tangential; then prepare two discs of metal, of the same radii as these arcs, that can be held by friction on the axis of the anchor. When this also is fixed on its axis, it will be easy, by placing a perfectly straight rule against each incline in succession, to ascertain whether its direction is satisfactory or otherwise; observing whether the rule, when passing through the point *o*, coincides with the tangent to the corresponding disc. Of course allowance must be made for any minute excess of metal left as a precaution, to be removed after either verifying the escapement or smoothing and polishing the pallets (**933**).

1002.—The pin escapement is advantageous in that it does not involve such accurate workmanship as the Graham, and it is not liable to catch when the pivot-holes of the anchor have worn larger.

If the pins are sufficiently long (without the risk of bending), somewhat enlarged, if possible, at their summits, and the acting surfaces of the pallets rounded in a beaded form so as to retain the oil by capillarity (**81**), this form of escapement is not liable to wear and it retains the oil for a long period (**1009**). These advantages have led to its very general adoption for turret clocks.

The anchor should always hang vertically, as in fig. 6, plate XII., in order that the oil may remain on the pallets.

Experiments have been made on the effect of facing the pins with various alloys, and with vegetable or animal substances, such as horn, etc.; although some of these attempts have been fairly successful and although the hardened steel pins, provided with oil, work well, manufacturers have always come back to pins made of good brass; the other materials that have been tried either exhibit objections in time or do not possess sufficient recommendations to make them preferable.

In turret clocks, with a view to increased facility of construction and adjustment, the anchor is often made of brass and fitted with hardened steel pallets; and, in chimney-piece regulators, the anchor is most usually formed of brass and fitted with jewel pallets, but they are also sometimes met with entirely of steel.

With a view to increase the rigidity, it is a common practice to form the under sides of the pallets as shown by dotted lines at *c*, fig. 8, plate XI.

1003.—*Details of a pin escapement*, which had an excellent rate and formed part of a clock by M. Borrel:

Diameter of escape-wheel, 80 mm. (3·15 ins.),—60 pins.

Length of escapement arms, 40 mm. (1·58 ins.).

Lift on each pallet, 2°:—1° for the pallet and 1° for the pin.

Supplementary arc on each side, 1°·5 (1° 30′); thus the total arc is 5°.

Pendulum: weight, 12 kilogr. (26·5 lbs.); beating seconds.

Comparing millimetres and grammes, the length of the escapement arm is to

The weight of pendulum ... : : 1 : 300
and to the virtual length ... : : 1 : 25.

The impulse was applied through a remontoir.

Pin Escapement with movable arms.

1004.—In this arrangement the only modification consists in the two arms of the anchor being movable with friction on their axis, and they are drawn together by a spiral spring, as indicated in fig. 15 of plate XII. This device is intended as a precaution against the pins being in any way damaged if the pendulum oscillates when the motive force has run down.

It would appear that the spring should not be too stiff, for it would then be without effect; nor too elastic, as it would interfere with the action of the escapement. Its exact strength will be determined by the resistance opposed by the pin to distortion.

Pin Escapement with pallets attached to the pendulum rod.

1005.—Influenced by the motives that led M. Winnerl to devise the escapement described in article **992**, M. Vérité constructed, as early as 1832, turret clocks in which the pallets were simply attached to the pendulum rod; the ordinary pin escape-wheel was used. Fig. 8, plate XI., does not require explanation.

In an escapement of this form the upper block *m* attached to the suspension spring should be carried by a slide, which, by means of a fine screw, can be moved to the right or left in order to set the escapement in beat. Or this block may be carried by a screw supported between two bearings in a horizontal position.

1006.—*Details of a pin escapement without the crutch*, by M. Vérité. (Fig. 8 is, however, not drawn in accordance with these dimensions);

Diameter of wheel, 80 mm. (3·15 ins.).—45 pins.

Length of escapement arm, 73 mm. (2·87 ins.).

Mean total arc, 4°; mean supplementary arc on either side, 1°.

Pendulum; weight, 7·5 kilogr. (16·5 lbs.); beating seconds.

Comparing millimetres and grammes, the length of escapement arm is to:

The weight of the pendulum about	: : 1 : 103
and to the virtual length	: : 1 : 14.

Certain other forms of Pin Escapement.

1007.—In the *first*, of which we shall only say a few words, the anchor is inverted and attached to a prolongation of the pendulum (carried on a knife-edge) above its point of support. The crutch is thus suppressed. We have seen an arrangement such as this in a clock made by the firm of Lepaute about the year 1820.

In the *second* form the anchor is placed on the opposite side of the wheel providing the motion of this latter is in the same direction as in fig. 6 of plate XII., or on the same side if the wheel turn in the reverse direction; it thus follows that the impulse planes must be formed as shown in fig. 7, plate XI. Instead of the wheel exerting a drag on the pivots of the anchor, it thus has a tendency to raise them and the resistance due to friction is slightly diminished. But the advantage gained by this small economy of force, indicated by a somewhat greater supplementary arc when the clock is first set in action, is neutralized by the necessity of the pivots accurately fitting the pivot-holes in order to prevent them from shaking, and the principal advantage of the ordinary pin escapement is thereby

lost. This advantage consists in the fact that there is no danger of catching through the enlargement of the pivot-holes (**1002**), and the effect of the oil is not detrimental; whereas the influence of an insufficient play is sensible.

This form of escapement, which may occasionally be of service, was formerly employed by a member of the firm of Lepaute; he made a narrow channel along the faces of the pallet, so arranging that the oil always returned along the inclines.

We shall do no more than refer to a *third* kind of this escapement in which the pins are planted so as to lie in the plane of the wheel, projecting outwards; this plane is perpendicular to that of the anchor, and the wheel is alternately pressed to opposite sides in the direction of its axis; the inconveniences of such an action are obvious.

Vulliamy's pin escapement with movable arms.

1008.—This escapement, of which the anchor is represented in fig. 11, plate XI., has been used for colossal turret clocks. As the pallets are called upon to support very considerable pressure, the inventor made their faces flat and thus arranged that the pins should be in contact throughout the entire width of the pallet.

With a view to realize this condition, each pallet, *l*, is prolonged by a cylindrical piece, *c*, which is adjusted in the piece *p p*, carried by the solid frame *a a a*, in such a manner that it can turn through an arc of about 3°. It results from this slight movement being possible that, even if a pin is not exactly perpendicular to the surface of the wheel, when it presses against the pallet the resting face will be brought parallel to the axis of the pin, if not already so.

To produce a similar effect when the pin acts against the impulse face, the piece *p p*, which carries the pallet, is mounted between the screws *v*, *v'*, and can rotate through a short arc round the axis *v v'*.

The spring *r* is traversed by the cylindrical piece *c* and pressed upon by the screw *e*, so that the pallet *l* is kept in contact with the piece *p p*.

1009.—*Details of a pin escapement by Vulliamy.*—(Taken from M. H. Robert's *Etudes sur l'horlogerie.*)

Wheel had sixty pins; it made one rotation in 4 minutes.

Arc of oscillation, 6°,—3° on either side of 0°.

Mean length of the escapement arm, 120 mm. (4·72 ins.).

Pendulum: weight, 67 kilogr. (147 lbs.); virtual length, 3·975 metres (13 ft ½ in).

Comparing millimetres and grammes, the length of escapement arm is to

The weight of the pendulum about ... : : 1 : 558
and to the virtual length ... ... : : 1 : 33.

Observation.—Vulliamy's clock was not provided with a remontoir; its rate was remarkably good and this fact is due to two principal causes: the length and weight of the escapement arms and pendulum were well proportioned, so that this latter was very little affected by variations in the motive force, and all the parts were made with great care; the acting surfaces were thus maintained in excellent condition, but such care could only be taken with the most expensive clocks. Thus, notwithstanding its success, this is not a clock to be copied except when very heavy hands require to be moved. Increasing the mass that has to be set in motion involves an unnecessarily great motive force, and it is quite possible to ensure a very satisfactory rate at less cost and with a more simple escapement.

Locking and impulse faces that are more or less rounded crosswise according to the pressure they are subjected to, will not wear until the oil required by the escapement is entirely dried up; for this oil is constantly being brought back to the acting faces by capillarity (**81**), whereas, when a pin acts against a flat face, a great portion of the oil is forced from the acting face, and, as there is no attracting force to bring it back, only a thin layer is left, which dries up rapidly; hence the acting parts are worn if not frequently oiled.

RÉSUMÉ OF SOME OF THE PRECEDING PARAGRAPHS and experimental data designed to facilitate the determination of the proportions best suited to any given clock escapement.

1010.—*The lifting angle as compared with the length of pendulum.* Details relating to the (anchor) escapements of three clocks that possessed good rates:

Graham Escapement, beating seconds.—*Lift* 1°, length of pend.: 994 mm. (39·1 ins.).

Graham Escapement, beating half-seconds.—*Lift* 2°, length of pend.: 248 mm. (9·76 ins.).

Ordinary Anchor Escapement, beating third-seconds.—*Lift* 8°, length of pend.: 110 mm. (4·33 ins.).

The lengths of pendulums are : : 400 : 100 : 44.
The Lifts are : : 50 : 100 : 400.

It is impossible, with the means at our disposal, to ascertain

with sufficient accuracy the mechanical value of the lift in each case. At the same time we feel satisfied that if they could be determined, we should find the second of these proportions to be the exact converse of the first.

While indicating this curious result with considerable reserve, since the question of the lift is complex, we would observe that the third escapement had a silk suspension and the difference observed was, in all probability, attributable to this fact.

Comparing the lifts in the several escapements described above, we observe that the Brocot and pin escapements for the same length of pendulum require rather more lift than Graham's, and this fact can be explained: for the pin escapement is often used under conditions that render an increased supplementary arc necessary, and the lift in Brocot escapements is succeeded by an impact of greater energy; there is thus a loss of power which must be renewed.

1011.—*The length of the escapement arm* cannot be measured by the lifting angle or the arc traversed (that is to say, as the inverse of this arc), for, with a given pendulum and motive force, the action on the impulse arm is very little changed if the length of this arm is varied while the lifting angle remains the same.

As to seeking a direct proportion between the length of these arms and the height of impulse planes, it need only be pointed out that, if such a proceeding were admitted, it would be necessary to adopt a length of arm that was constant, or nearly so, for wheels that differed considerably in diameter while the intervals between the teeth remained the same.

It seems needless to discuss this point further.

1012.—*Various proportions that have been adopted between the escapement arm and the virtual length of the pendulum.* (In all cases the pallets communicated the impulse to the pendulum through the intervention of a crutch).*

Graham and Pin Escapements	: : 1 : 25
Brocot Escapement	: : 1 : 14
Gable „	: : 1 : 15
Small ordinary anchor „ with heavy pendulum	: : 1 : 30
„ „ „ with light pendulum	: : 1 : 40

* The ratios (already given in the case of several forms of escapement) between the length of escapement arm and pendulum are not strictly comparable unless we assume the length of crutch to bear an invariable proportion to that of the pendulum, whereas it varies with different makers.

The above numbers are approximate and will facilitate the determination of better proportions; but there always remain to be ascertained by the sole method available at the present day, namely the experimental, the most convenient dimensions for the escapement arm as regards the resolution of force on the lifting face, the momentum of the pendulum, resistance of the suspension, length of crutch, etc. (**1014-5**).

1013.—*The supplementary arc* should be very limited; as a rule this is a condition that must be satisfied in regulators in order to secure a good rate. It has somewhat greater extent in clocks for ordinary use, when there is any occasion to increase the momentum of the pendulum, and the force transmitted to the escapement is likely to vary; whether such variation be due to bad depths, irregularity in the action of the motor, or to the number of parts that the clock is called upon to move, etc.; but this correcting effect must be applied with discretion since it involves some inconveniences (**933**).

1014.—*Length of crutch: consequence of its being suppressed that has not before been noticed.* It is not a matter of indifference whether the pendulum act in conjunction with a long or a short crutch. By lengthening this crutch we increase the power-arm at the same time that the resistance-arm, measured along the pendulum, is increased in the same proportion; thus, from a geometrical point of view, we need take no account of anything but the increased weight of the crutch; we should, however, commit a grave error if the question were thus only regarded from one point of view.

The increase in weight is, in certain cases, of less importance than the *choice of the point at which the force is applied* along the pendulum rod. The problem to be considered is precisely the same as is discussed in **100** and the following paragraphs. With a given pendulum, in proportion as it is struck nearer the centre of percussion, it becomes more and more sensitive to variations in the motive force, but the effect of this force on the suspension-spring becomes gradually less; hence it follows that reducing the length of crutch (if we ignore the effect of variations in the friction, shake in the crutch, the amount of which depends on the length of pendulum, etc.) produces the same result as we endeavoured to secure by a definite reduction in the length of escapement arm; a result which was not in reality secured

unless *the crutch was such as to correspond* to this reduced length. In the details given in article **1006**, the ratio :: 1 : 14 is not directly comparable with proportions taken from an escapement with the crutch; it implies a greater difference than the proportion :: 1 : 25 would in the case of these latter. This point is here raised for the first time and well deserves attention and careful investigation at the hands of intelligent horologists.

In ordinary practice the mean length of the crutch is usually said to be between a quarter and a third the length of pendulum (or one fifth in the case of a seconds pendulum). With a view to avoid the detrimental effects due to the play of pivots, the variation of oil, shake of the pendulum rod in the crutch fork, etc., which are especially sensible with a short crutch, it becomes necessary to make their length considerable when small pendulums are employed. The timing would be better if these pendulums received the impulse in a more satisfactory manner (**1297**).

1015.—*Choice of a pendulum.*—The length depends on the number of oscillations that it beats in an hour. If the length of the escapement arms is fixed, it may happen that with the given number of oscillations the clock can never be timed, and it becomes necessary to select another.

General Rules.—I. *The length of escapement arm* (with suitable crutch) *is determined by the point at which the pendulum is sufficiently insensible to variations in the motive force, and conversely;* —II. *The weight of the pendulum must be ascertained from the time it requires* (after being displaced from the vertical by an angle equal to the lift) *to attain to its maximum regular oscillation.* This novel principle, which we have introduced into the practice of horology (**440**), will give an easy means of determining the weights of pendulums; for it will only be necessary to cause the lifting angle, the supplementary arc, weight of bob, etc., to vary, until the whole is so correlated that the pendulum does not attain to its full uniform arc either too slowly (as happens with a heavy pendulum or a feeble motive force); nor too promptly, as this would point to the pendulum being too light or the motive force excessive.

We shall do no more than give this outline of the main principles, which must be supplemented by the study of well-proportioned clocks; for as yet we are not in possession of a

sufficient number of actual observations to corroborate results obtained by calculation.

1016.—The usual weights of the pendulums employed have been indicated under each escapement. These figures will serve as a guide in practice, but they must always be considered subordinate to the principles already laid down.

REMARKS.—A change in the weight of the pendulum bob alters the relation subsisting between the lifting and supplementary arcs, between the length and period of the oscillation, etc.; and thus all the proportions are modified. The changes may be such that one neutralizes the other and the rate remains unaltered; or, one influence being in excess, the rate will be changed for better or worse; either immediately or in the course of time, etc.

Virtual or (approximately) *Simple Pendulum.*—In speaking of the length of pendulum, it is always the virtual length that is considered. This may be experimentally ascertained by balancing the pendulum horizontally on a knife-edge; the interval between the fulcrum and the extremity of the rod added to the length of suspension-spring gives the required dimension. This method is sufficiently exact when the rod is light and not much loaded at its upper end; but when such is not the case it frequently gives a very erroneous amount and varies according to the nature of the suspension.

1017.—*Number of teeth and diameters of escape-wheels employed in modern clocks.*

The three sizes of movement most usually employed are

				mm.			mm.		
That known as the	3-in.	movement		(81)	with escape-wheel		19	(0·75in.)	in diameter
„ „ „	$3\frac{1}{4}$ „	„		(88)	„	„	20	(0·79 „)	„
„ „ „	$3\frac{3}{4}$ „	„		(101)	„	„	22	(0·87 „)	„

There is thus an increase of 1 mm. to correspond to each quarter of an inch.

In these movements, when the escape-wheel performs 120 revolutions in an hour it has

30	teeth	with a	virtual	pendulum	of	248	mm.	(9·76 ins.)
32	„	„	„	„	„	219	„	(8·62 „)
34	„	„	„	„	„	194	„	(7·64 „)
36	„	„	„	„	„	171	„	(6·73 „)
38	„	„	„	„	„	153	„	(6·02 „)
40	„	„	„	„	„	140	„	(5·51 „)
42	„	„	„	„	„	126	„	(4·96 „)
44	„	„	„	„	„	117	„	(4·61 „)
46	„	„	„	„	„	106	„	(4·17 „)
48	„	„	„	„	„	97	„	(3·82 „)

1018.—These experimental data are given merely as a help to facilitate adjustment; for, with a given size of escape-wheel,

the length of the pallet-arm is dependent on the length of pendulum (**1012**), and if, on the other hand, the pendulum or, in other words, the number of vibrations per hour, is fixed, the size of the escape-wheel must be deduced from the length of pallet-arms, which will depend on the number of teeth or the arc covered by the anchor. It may be determined either geometrically or by calculation, taking as a basis the dimensions met with in an escapement that has proved to be satisfactory.

1019.—*Arc of the circumference of the wheel covered by the anchor*, in clocks of modern construction; no account being taken of the number of teeth.

Large anchor of regulators between	-	-	90° and 120°.
English anchor - - „	-	-	69° and 90°.
Ordinary small anchor - „	-	-	40° and 60°.

These figures show a considerable latitude, since the arc varies with the length of the escapement arm. They will do no more than afford the basis of a first approximation to a maker desirous of designing a model escapement.

CONCLUDING OBSERVATIONS.

1020.—The very important part played by the suspension of a pendulum renders a special article on this subject necessary; it will be found in the Third Part of the work (**1299**).

1021.—It is hardly necessary to remind the reader that:—the centre of flexure must lie on the prolongation of the axis of rotation of the anchor;—a spring suspension must be gripped firmly and without play in the slide;—the pendulum rod must neither be held too tight in the crutch-fork nor shake in it; any tendency to shake in a pendulum bob resting on a nut must be prevented; the suspension hook that carries the pendulum should have an obtuse angle and not be semi-circular, in order that it may hold more firmly and without a possibility of lateral displacement;—and, lastly, it is well to slightly round off sideways the escape-wheel teeth, finishing them with soft charcoal, etc., etc.: and we will conclude with this one observation; success in the construction of clocks as well as of watches always depends on the arrangement as a whole; a study of the theory, supplemented by the very numerous experimental data accessible at the present day, will, after a few trials, infallibly lead to success; but the uniformity thus attained to can only be assured by *very careful workmanship*, and this alone can prevent the variations of the motive force from exceeding the limits beyond which the escapement is incapable of maintaining the rate constant.

NOTES

ON VARIOUS CLOCK ESCAPEMENTS.

1022.—ESCAPEMENT OF M. DESHAYS, *also known as the crank escapement.*—This is represented in fig. 11 of plate XIII.; it first appeared at the Exposition of 1827. With the exception of a slight difference in form, it is identical with that exhibited at the International Exhibition of 1851 by MacDowall, and described in articles **893-4**. A crank escapement without locking faces was attempted by the Abbé Soumille in 1747. (Both the crank and pin are drawn too large in fig. 11.)

1023.—ESCAPEMENT OF M. P. GARNIER.—A A and B B (fig. 12, plate XIII.) are two wheels mounted on a single axis: *c c* is a semicircular disc fixed to the axis of the balance or pendulum; its diameter presents two edges rounded like those of a cylinder. The point of each tooth is locked against the flat of *c c*, and the impulse results from the incline of a tooth pressing against one of the edges.

This is a novel and legitimate combination of several of the features of the cylinder escapement and those of Enderlin and de Baufre. It was intended mainly for use in small portable or carriage clocks, the manufacture of which in Paris is much indebted to the energy of M. Garnier. The low price now charged for cylinder and lever escapements has prevented it from coming much into use.

1024.—SINGLE BEAT ESCAPEMENT *with hinged pallet.*—With a 30-tooth wheel, an anchor or pin escapement and a pendulum of 994 mm. (39·1 ins.) a hand carried on the escape-wheel axis will beat seconds. In order that it might turn once in a minute with a 248 mm. (9·76 ins.) pendulum, we should require a 60-tooth or pin wheel, and this would require the teeth to be very close together, or the wheel to be too large in proportion to the train. To avoid these two inconveniences what are known as single-beat, divided pallet, or deer's-foot escapements have been introduced; their effect is to make each alternate vibration dumb so that with a 30-tooth wheel the hand will mark seconds. Fig. 9 of plate XIII. represents one of the most simple and usual forms of these escapements.

A B is the anchor for a pin escape-wheel. The arm A is of

the ordinary construction. B carries a pallet *h o*, that is cradled or movable on pivots and always has a tendency to rise, through the excess of weight at *h*. This free pallet is held against the fixed pallet *n* by the pressure of a pin as indicated at *o*; with the movement of the anchor this pin passes from the pallet *o* to *s*. The extremity of *o* is immediately raised until *h o* occupies the position *x x'* and, on the return of the pendulum, the pin is thus allowed to escape; it passes over the impulse face of *n* below the pivoted pallet, and this latter is again brought by the succeeding pin into the position shown in the figure. A hand mounted on the axis of the wheel will therefore only move once for each two oscillations of the pendulum.

Observation.—None of the escapements with divided or hinged pallets, whether working with a weight or spring, have given good results, although several have been made varying in form; and this fact is due to the effects of adhesion, the stickiness or clogging of the oil, by which they are always influenced in time.

1025.—OTHER SINGLE-BEAT ESCAPEMENTS BEATING SECONDS.—One of these is represented in fig. 7, plate XIII., and its mode of action is as follows. A locking occurs at *a*. After the unlocking the tooth *a* advances slightly, and the pin *n* rests against the second locking face *c*. The pendulum in returning releases it, and the impulse is effected by the pin *n* pressing against the concave incline *h*; it terminates when the tooth *b* is locked at *a*, etc.

Although simplicity and certainty of action render this escapement preferable to those with hinged pallets, it cannot be expected to secure a regularity equal to that of the anchor escapement of regulators.

Observation.—A great variety of escapements, both of the frictional rest, detached and single-beat class, have been designed with a view to beat seconds when the pendulum measures only 248 mm. (9·76 ins.); but the results have not been satisfactory. It is better to give up the attempt as useless and to employ the ordinary escapement giving two beats in a second. When necessary there is no difficulty in counting every alternate beat.

1026.—GRAVITY ESCAPEMENTS.—Under this term are included a variety of escapements in which the impulse is applied to the pendulum by a body which acts solely in virtue of its weight,

that is to say of the laws of gravitation. Although a considerable number of escapements based on this principle have been designed, we shall only describe one proposed by Denison.

1027.—*Denison's Gravity Escapement.*—Fig. 6, plate XI., represents one form of this escapement. Two pallet arms T and R are pivoted at their upper ends, the impulse being applied by the pins *i*, *c*, *u*, acting against the extremities of the inner arms *s*, *k*. The figure shows the positions of the several parts as soon as the pallet T has communicated an impulse to the pendulum, which continues its motion towards the right until the head of the screw *m* comes in contact with a pin at the lower extremity of the pallet R. The resting arm or leg *k*, is then released and the wheel turns, impelling the pallet-arm T to T *z*, by the pin *c*: the motion of the escape-wheel is arrested by the leg *a* resting against the pallet *f*. The three legs are thus brought into the positions *f*, *k*, *j*; and, in the return oscillation of the pendulum, R is carried with it until arrested when in the vertical position: the requisite impulse is thus applied.

The pallet-arm T is now in a position corresponding to that of R in the figure and a similar action to that above explained will take place in the opposite direction.

1028.—The pendulum never comes in contact with a pallet-arm until it has been moved from its first to the second resting position; the duration of the impulse depends on the interval between the dotted line *z* and the arm T, and its amount varies with the weight of this arm. A fly, *v v*, mounted on the axis of the escape-wheel, reduces the impact on the locking pallets.

The escape-wheel has been made with four or five legs. The double three-legged escapement has, however, been generally preferred,* and an escapement of this construction has secured an almost absolutely perfect rate in a turret clock of very large dimensions.

* Readers that require further information on the subject of Gravity Escapements should consult Denison's work on Clocks, watches, and bells (Weale's Series).

PART II.

DEPTHS.

Reply to certain criticisms.

1029.—In the first edition of this work the arrangement adopted was as follows: Escapements, depths, motors and miscellaneous articles. We have retained this order since experience has proved it to be logical. A Treatise on Horology of the extent and character of this work is mainly intended for the use of watchmakers who are already competent to make at any rate some parts of a watch or clock; so that it is absolutely necessary that he know both the mode of action of the several escapements, and the amount of variation in the motive force they will admit of, while continuing to exercise their controlling or regulating faculty. It is only when the reader is possessed of this knowledge that he can advantageously study the manner in which force is transmitted and examine the various sources of power.

If we were writing a purely elementary or descriptive treatise an entirely different course should be pursued: for in such a case it is well to follow a system similar to that through which every apprentice should pass, gradually advancing from what is easy to what is more difficult: from the barrel to the train and from the train to the escapement; the case is entirely different, since the student is then merely required to make with his own hands a certain number of pieces, copying definite models.

Preliminary considerations on depths.

1030.—Generally speaking, by the term depth we understand a system of two wheels, one of which communicates motion to the other by means of projecting pieces or teeth formed on the edge of the two circles, the projections of one circle passing into the hollows of the other. By the term *pitch* is understood the sum of a *space* and a *tooth*.

For distinction the larger of the two is generally known as a *wheel*, and the smaller, if it have less than twenty teeth, is called a *pinion*. The teeth of the larger wheel retain the name "teeth," while those of the pinion are called *leaves*. The solid centre of a pinion is known as its *heart*.

A piece that moves on an axis is termed a *mobile* (**17**), a name however that more especially refers to a wheel and pinion.

1031.—The invention of toothed gearing is of the highest antiquity: it was employed by the Egyptians, the Greeks and the Romans; but it is not known to what degree of perfection they arrived. The Egyptians are generally considered to have been the first to employ it in machines for the measurement of time, and, judging from the descriptions of complicated machines that historians have handed down to us, the Arabians appear to have, from a very early period, possessed considerable practical knowledge of this subject.

The middle ages, up to the time of the Commonwealth, do not seem to have introduced any special improvements in this important branch of Mechanics. As evidence of this fact it is enough to examine gearing of this period, when it will be seen that the forms of the teeth are very varied; they show that the watchmakers did not follow any definite system in forming their depths. Some adopted particular forms after a long course of experiment and very many attempts; others copied, more or less successfully, teeth that had been proved by experience to be satisfactory. On the whole their trains were very defective, not going at all unless the impelling force was excessive, and indicating the time with great irregularity. It was this imperfection of the depths that suggested to Huyghens the idea of a remontoir.

Two savants, Roëmur and Lahire, the first a Dane and the second a Frenchman, were the first to perceive, about the middle of the 17th century, that science was able to take cognizance of the subject of depths and to lay down accurate principles for their construction.

Camus, a member of the French Academy of Sciences, published a memoir on the subject in Thiout's *Traité* in 1741, and some years later he gave in his *Cours de Mathématiques* the most complete treatise we possess on this subject.

F. Berthoud and Le Roy demonstrated experimentally the excellence of the laws given by theory; and Lalande considered the subject of depths in an exact manner, but without the use of high mathematics, in Lepaute's work published in 1767.

Moinet* has, more recently, enriched the theory of depths

* Louis Moinet, who was born at Bourges in 1768 and died in 1853, was a highly intelligent and distinguished professor of the arts, a very clever amateur in horology and a skilful writer; at the age of 75 he published a *Traité d'horlogerie*. It is

by the publication of several practical methods that satisfy the requirements of theory, and, in England, Airy and Willis have considered the subject both from a theoretical and practical point of view.

1032.—It was not our original intention to consider the subject of depths in this work. For what could we say about it that has not already been said far better by others? But a long experience of watchmakers has convinced us that, from their point of view, some authors are incomplete, others too fond of geometrical formulæ, while others are so wanting in method that the reader soon becomes wearied and gives up the study; for it must be remembered that a workman is not accustomed to study and has but little time to devote to it. Nearly all these authors, moreover, abstain from giving practical details and methods, although these are so useful and so scarce and are especially inaccessible to young watchmakers. The practical portion will therefore be brought into special prominence in the following pages; as to the theory, we will confine ourselves to simplifying and explaining as clearly as possible the principles that have been so fortunately discovered by the authors above referred to.

INTRODUCTION

TO THE STUDY OF DEPTHS.

1033.—As the action of a depth is in reality nothing more than that of a series of levers succeeding each other without appreciable intervals, we must remind the reader that the general theory of the lever can be summarized as follows (**56**): *in every lever, wherever the fulcrum be situated, the power is to the resistance in the inverse ratio of the lengths of their arms;* and we would refer him to the *Elements of applied mechanics* at the commencement of the work, which are rendered more complete by the several articles that follow, relating to the bent lever, friction, and the mathematical curves that are most frequently employed in depths.

The Bent lever.

1034.—When a lever is not a straight rod, the actual length of the arms does not accurately indicate the ratio of the power

much to be regretted that this in many ways excellent work, which however was undertaken too late in life, was marred by the infirmity of its author and perhaps also by the very serious difficulties that stood in the way of its publication.

to the resistance. This ratio may be determined by comparing the *virtual* length of the lever arms.

If the direction of the force is known, the virtual length is the length of a line drawn from the fulcrum perpendicular to this direction.

Thus if the direction of the forces applied to the arms A R C, A R N (fig. 64) of two bent levers be indicated by the lines C S, N C,

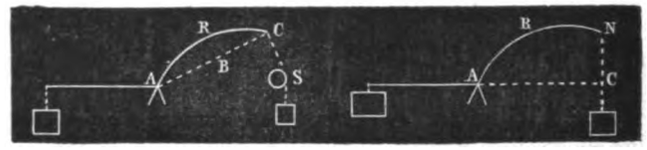

Fig. 64.

the powers of the levers are compared by reference not to the arms A R C, A R N, but to the *virtual* arms A B C, A C.

Friction at Depths.

1035.—Paragraph **105** draws a distinction between the character of the friction according as the point of contact of the two levers is advancing towards the line of centres or receding from it; the two kinds are called respectively *engaging* and *disengaging* friction. The following practical explanation has been given of the effects produced by the two kinds:

All bodies, however highly they be polished, are covered with innumerable minute irregularities, due to the pores that exist between the molecules constituting the body and the manner in which these molecules arrange themselves in accordance with their chemical affinities. This roughness, although quite insensible to the touch, is yet visible in the great majority of bodies by the aid of optical instruments; it can never be perfectly removed in consequence of the imperfectness of our means of polishing.

When the two surfaces are pressed one against the other the asperities of the one engage with those of the other, and thus a resistance is opposed to their motion when in contact. If this sliding action be caused to take place several times in succession, a fine dust or powder is found between the two surfaces, resulting from the breaking off of projecting points, and it is evidence that wear has commenced.

This being granted, assume A and B (fig. 8, plate XIV.) to

be two teeth, the asperities of which have been purposely exaggerated in order to illustrate our explanations. If the tooth A impel B while moving away from the line of centres, in other words towards the right, the action will not be opposed by any great resistance; but such is not the case when B is impelling A towards the line of centres, that is, to the left: for then the roughnesses will butt against each other, and, in order that the movement may continue, it will be necessary that they be stripped off; a much greater force will therefore be required than when the movement takes place in the opposite direction. Moreover, however fine the irregularities be, there must be more rapid wear in the second case than in the first, since at each engagement the projections are thus stripped off. Everyone knows how different is the force required to draw a body along from that necessary to push it with an inclined handle.

The effect under consideration is strictly comparable on a small scale with that of a ratchet wheel and side spring. The former will move easily when turned in one direction but cannot be rotated in the opposite direction except by stripping off the teeth in succession.

1036.—We had intended to discuss the question of friction in some detail, but this would occupy too much space to be in keeping with the rest of the work, and we are therefore compelled to merely summarize this important subject.

If metallic surfaces be examined by the aid of a microscope of considerable magnifying power, it will be seen that they do not all present the same appearance; the texture of some seems to resemble a collection of small spherical molecules, while others seem to be covered with a number of flaws. Hence they differ as regards friction, which varies, not only with the molecular nature of the surfaces, but also according as their movement is continuous or intermittent; for the condition of the surfaces, as depending on the heat and pressure due to friction, will be modified if there is a period of rest.

It is in part to this cause that we must attribute the change observed in the rate of a chronometer that has not been going for some time.

We would direct the attention of our more skilful fellow-workers to these delicate phenomena; and, if our very numerous duties permit of it, we hope to revert in a subsequent work to the subject, which is so full of interest.

On the curves that are most usually employed in depths.

THE CYCLOID, EPICYCLOID AND HYPOCYCLOID.

1037.—If a circle of wood, metal or cardboard be caused to roll, without slipping, along a straight rule, the circle carrying a point of a pencil at the point *a* (fig 8, plate XIII.), where the circle and rule are, in the first instance, in contact, this *generating point*, *a*, will trace out, on a sheet of paper previously placed underneath, the curve *a b c*.

This curve, known as a *cycloid*, is, then, the curve generated by any given point in a circle when rolling along a straight line.

1038.—If a circle be caused to roll on the external side of the circumference of another circle, the generating point *e* (fig. 18, plate XIII.) will trace out the curve *e f g*, which is known as an *epicycloid;* and if it roll in the inside, the curve will have the form *m n p*, termed a *hypocycloid.*

Those who are not acquainted with geometry may obtain a clear idea of the manner in which these several curves are generated by cutting out discs and circular holes in cardboard, and moving them in the manner indicated on a sheet of white paper.

1039.—*Properties of these curves.*—One very remarkable property is common to all three curves. A straight line passing through the generating point, *o* (fig. 18), and the point at which the circle and straight line (or the two circles) touch is always perpendicular to the curve, that is to its tangent.

As will be shown in article **1087**, the advantages secured by forming the teeth of wheels to correspond to these curves are in great part due to this important property.

THE INVOLUTE OF A CIRCLE.

1040.—The involute of a circle is a plane curve generated by a point on a straight rule adjusted to move, without slipping, round a circle, to which it always remains tangential.

Suppose a thread to be wound round a fixed circle, and, holding this string by its extremity *a* (fig. 3, plate XIII.), unroll it, always keeping the string stretched; the point *a* will pass through a', a'', a''', etc., and will trace out the involute, for a stretched string will satisfy all the conditions of a straight rule moving as above explained.

The normals (or perpendiculars) to the involute of a circle are always tangents to the fixed circle.

THE HELIX.

1041.—The helix is a curve of double curvature generated by the point of contact of a straight incline when moving round a cylinder.

Let *b c f d* (fig. 1, plate XIII.) be a rectangular sheet of some flexible substance. If, after having drawn on it the inclines *c*, *n*, *n*, *n*, the sheet be coiled round a cylinder of the same height, the length of whose circumference is equal to *d f*, the inclined planes will trace out a gradually ascending continuous curve, *n' n' n'*, on the cylinder. This curve is known as a *helix*.

The ordinary screw is based on this principle of the inclined plane; the first drawing (fig. 1), with only one line traversing the cylinder, corresponds to a single threaded screw. The *pitch* of this screw is the total height of a hollow and thread.

If a number of lines, such as *z*, *x*, *c* (fig. 1), or *g*, *h*, *i*, *j* (fig. 2), divide the width of the rectangle instead of its height into equal parts, we shall obtain a screw with several threads or the pinion of a helical depth. The *pitch* of a screw with several threads is the height through which any given thread ascends with one turn of the screw.

The primitive or geometrical circles and diameters.

The points or curves of teeth.

1042.—In every depth the curved portions, both of the leaves and teeth, which are known as the *points* or *curves*, always project beyond the two circles A and B (fig. 4, plate XIII.) that are tangential to one another at the point *o* on the line of centres. In discussing any depth we start with the supposition that if these two circles were to roll on each other without friction the depth would be perfect or primitive. Hence they are known as the *primitive*, *geometrical* or *pitch circles*, and their diameters and radii are also called geometrical or primitive.

1043.—Thus in every wheel or pinion it is important to remember that the total diameter is the primitive diameter plus twice the height of the point.

As the uniformity of the lead depends mainly on the determination of these pitch circles, we would warn our readers that a knowledge of the two diameters, the primitive and total, and the careful distinction of the one from the other is of the

greatest importance, not solely in order to understand the theory of depths but also to make useful practical applications of this theory.

The point of a tooth is occasionally termed the *ogive*, while that of a leaf is called the *rounding*, on account of the forms they generally have in horological mechanism.

CALCULATION OF THE VELOCITIES OF MOVEMENT OF THE SEVERAL PARTS OF A TRAIN.

To which is added the calculation of the number of vibrations of a balance.

1044.—In this portion of our discussion of the theory we propose to consider the relative number of rotations of the several mobiles and, as a necessary consequence of these proportions, the number of teeth and leaves of the wheels and pinions; subjects which must, of course, be examined before we proceed to ascertain what are the best forms to give to the acting surfaces.

Ratio of the primitive diameter to the number of teeth.

1045.—If one wheel drive another by the mere contact of their two circumferences (fig. 20, plate XIII.), it is not difficult to see that, if there be no slipping, each portion of the driving wheel will advance an equal portion of the circumference of that driven, so that, if this latter has a circumference equal to half that of the former, the smaller wheel will rotate twice for each rotation of the larger. We know from geometry that the circumferences of circles are in proportion to their radii or diameters; hence it follows that, as the larger radius is to the smaller, so will the number of revolutions of the small wheel be to those of the larger. Thus the revolutions of the two wheels are *in the inverse ratio of their primitive diameters or pitch circles.*

1046.—As will have been seen, this method of transmitting motion is only a theoretical assumption, and it will be easily understood that, if the circumferences are not provided with teeth, irregularities due to slipping will occur. When the two wheels are thus indented it will be necessary, in order that the depth may give a uniform lead, that, throughout the entire period the tooth impels the leaf, each portion of the pitch circle

of the wheel should impel an equal length of the pitch circle of the pinion.

1047.—Now since a tooth and space of the wheel should occupy the same interval, measured along the pitch circle, as a leaf and space of the pinion do along its pitch circle, it necessarily follows that, whatever number of times the total number of these small arcs (each corresponding to a leaf and space) of the pinion pitch circle is contained in the primitive circumference of the wheel, so many rotations will the pinion make for each rotation of the wheel; hence it further follows that the small pitch circle is to the large one as the number of leaves is to the number of teeth.

It is somewhat difficult to measure the circumference; but, as we have already seen that the circumferences are to each other as their diameters, the one may be substituted for the other, and we conclude that:

When a wheel and pinion are pitched together, the primitive diameter of the wheel is to that of the pinion as the number of teeth is to the number of leaves.

Thus to secure a satisfactory depth between a wheel of 60 teeth and a pinion of 10 leaves, their primitive diameters should be in the ratio of 60 to 10, that is to say the primitive diameter of the wheel is 6 times that of the pinion.

1048.—This rule connecting the number of teeth and the primitive diameters is a fundamental law. If the proportion does not exist, it inevitably follows that a tooth and space on the circumference of the wheel will occupy a greater or less space than they do on that of the pinion, and no uniformity can be maintained in the lead, which will be accompanied by slipping, drops, catchings, etc.; for it may be well to point out, at the outset, that a pinion is not too large or too small unless its primitive diameter is at fault.

To determine the primitive diameters for a given depth.

Two main cases can occur and these we will present in the form of questions: either we know nothing but the distance between the two centres and the numbers of the two mobiles, or it is necessary to adapt a pinion with a given number of leaves to a wheel that is already made.

1049.—First Question.—*The distance between the centres of the wheel and pinion (measured on the calliper of the clock or watch)*

being known, as well as the numbers of teeth and leaves, how can we ascertain the length of each geometrical radius?

Let 60 mm. be the distance between the centres, the wheel having 32 teeth and the pinion 8 leaves.

Remembering the ratio that should exist between the primitive diameters and the numbers of the teeth, and observing that the 60 mm. is the sum of the two primitive radii, state the proportion as follows:

The two numbers taken together (32 + 8) are to the two radii taken together (60) as the greater number (32) is to the greater radius (x, that is to say an unknown quantity).*

The proportion then is, 40 : 60 : : 32 : x.

In every proportion the product of the extremes (40 × x) is equal to the product of the means (60 × 32), and if one of these products (60 × 30, or that in which both terms are known) be divided by the known term (40) of the other product, the value of x will be obtained.

$$60 \times 32 = 1920 \therefore x = \tfrac{1920}{40} = 48.$$

Thus 48 mm. is the geometrical radius of the wheel, and 60-48, or 12 mm. is that of the pinion. As we have seen, the primitive diameters will be obtained by doubling these radii, so that the diameters required are 96 and 24 mm. respectively (**1052**).

1050.—Another mode of determining the primitive radii is given in article **1103.**

1051.—SECOND QUESTION.—*A wheel being given ready made, as well as the number of leaves of a pinion, to find the geometrical diameter of this latter?*

Let the wheel have 50 teeth and a primitive diameter of 20 mm., and let it be required to pitch it with a pinion of 10 leaves.

As in the previous case, we shall have, in virtue of the proportion that subsists between the numbers of teeth and the primitive diameters;

50 : 10 : : 20 : x. $\quad 10 \times 20 = 200 \therefore x = 4.$

or the geometrical diameter of the pinion is 4 mm.

1052.—These examples will suffice; for whatever the problem under consideration may be, it always resolves itself

* We may take the two diameters or the two radii indifferently as is most convenient. For the radius of a circle is always half its diameter and thus the ratio remains the same.

into the determination of one term in the following proportion: The primitive diameter of the wheel is to that of the pinion as the number of teeth is to the number of leaves.

We would invite all watchmakers to accustom themselves to the theory of proportions, the study of which is by no means difficult. This theory is capable of affording immense assistance in the daily work of a watchmaker.

To calculate the velocities in a train of wheels.

Number of revolutions of the escape-wheel.

1053.—The solution of this problem consists in finding the velocity of any given wheel as compared with that of another, which is looked upon as the first: for instance, the velocity of the pinion of the escape-wheel, or the number of rotations of this wheel in an hour, that is, while the wheel that carries the minute-hand makes one revolution. The calculation will involve three wheels; the centre wheel and the two that follow and the three pinions that engage with them.

1054.—Before considering the proof we will give the general rule.

In order to ascertain the velocity of any given mobile in a train, *multiply together the number of teeth in all the wheels that precede it, and divide this product by the product of all the leaves of the pinions that engage with the several wheels; the quotient gives the number of revolutions of the last mobile that correspond to one revolution of the first.*

Let it be required to ascertain the number of rotations per hour of the escape-wheel in the following train of a watch.

Centre wheel, 70 teeth, engaging into a pinion of 10 leaves.
Third ,, 64 ,, ,, ,, 8 ,,
Fourth ,, 60 ,, ,, ,, 6 ,,
The last pinion being that of the escape-wheel.

The centre wheel carries the minute-hand and must therefore make one revolution per hour. It has 70 teeth and engages with a pinion of 10 leaves, so that this pinion makes $\frac{70}{10}$ or 7 revolutions in an hour; the third wheel, riveted to this pinion, of course makes the same number.

This third wheel has 64 teeth and engages with a pinion of 8 leaves, so the pinion makes $\frac{64}{8}$ or 8 revolutions for each one of the third wheel: but this latter makes 7 in an hour; there-

fore the pinion must make 7 × 8 or 56, as does also the fourth wheel which is carried by this pinion.

The fourth wheel has 60 teeth and, engaging with a pinion of 6 leaves, will cause it to make 10 revolutions for each one of the wheel. And the wheel makes 56; therefore the pinion makes 560 in an hour.

Thus the escape-wheel, carried by this pinion, makes 560 revolutions in an hour, that is while the centre wheel performs one revolution.

1055.—As will be easily seen, the preceding operation is nothing more than a series of divisions and multiplications, and a similar result would be arrived at by multiplying together the numbers of teeth of the wheels and dividing this product by that of the leaves of the three pinions engaging with them; the quotient would give the number of revolutions of the escape-wheel pinion and of the escape-wheel which is riveted to it. The first product will be found to be 268,800 and the second 480; the quotient therefore is 560, identical with that previously obtained.

To calculate the number of vibrations of the pendulum of a clock or balance of a watch.

1056.—When the number of revolutions of the escape-wheel is known it is easy to ascertain the number of vibrations, for we know that each tooth of the escape-wheel gives rise to two vibrations of the balance; it is only necessary, therefore, to multiply the number of revolutions by twice the number of teeth of this escape-wheel to obtain the required number.

Assume the wheel to have 14 teeth; we shall have 28 × 560 or 15,680 vibrations in an hour.

If it have 15 teeth, there will be 30 × 560 or 16,800 vibrations.

And with 16 teeth, 17,920 vibrations.

1057.—To sum up.—In order to determine the number of vibrations of a balance or pendulum per hour (that is, during each revolution of the wheel that carries the minute-hand), *multiply together the numbers of teeth of the wheels* from the minute-wheel inclusive, *and divide the resulting product by the product of the leaves of the pinions that engage with them.* As the quotient gives the number of revolutions of the escape-wheel per hour,

*it is then only necessary to multiply this number by twice the number of teeth in the escape-wheel, and we obtain the vibrations of the balance in an hour.**

1058.—The reader can practice himself by calculating the numbers in the following trains; they are very frequently adopted for watches.

Centre wheel.	Third wheel.	Fourth wheel.	Pinions.	Escape-wheel.	Vibrations of Balance.
80	60	54	10, 8, 6	15	16,200
76	64	54	10, 8, 6	15	16,416
78	66	60	10, 8, 6	15	19,805
68	60	58	8, 8, 6	15	18,487
64	60	60	8, 8, 6	15	18,000
68	60	56	8, 8, 6	15	17,850
56	50	50	6, 6, 6	14	18,148
56	52	52	6, 6, 6	14	19,628
58	52	52	6, 6, 6	14	20,330

1059.—*Clocks.*—All the preceding rules are equally applicable to the trains of clocks, which at the present day are usually so arranged as to give 120 revolutions of the escape-wheel in an hour. The number of its teeth is varied in accordance with the data contained in the table giving the lengths of pendulums that correspond to the varying number of vibrations per hour (page 814).

1060.—*Observations.*—When the product of the teeth is increased as compared with that of the leaves, the number of vibrations becomes greater. If then a watch cannot be regulated on account of its having an insufficient number of vibrations, it will be necessary either to replace one of the wheels of the train by another of higher number of teeth or to let it engage with a pinion of lower number, in other words, one that is smaller.

Thus in the example of article **1054**, if the centre wheel of 70 teeth be replaced by one of 75 and the 10-leaved pinion by one of 8 leaves we shall have 750 revolutions of the escape wheel in the same time as there were previously only 560.

It will of course be evident that such a change in a watch involves the use of a different balance-spring, and in a clock a change in the length of the pendulum.

* Other methods are employed to arrive at this same result, but, with the exception of a few special cases, they have no advantage over that given above: it is therefore unnecessary to consider them.

It is generally better to adopt the method here indicated rather than to increase either the diameter or number of teeth of the escape-wheel.

To calculate the time of going of a watch or clock, the number of teeth of a going barrel, &c.

1061.—Calculations of this kind do not present any difficulty; before laying down the mode of procedure a simple explanation is necessary.

It is necessary to commence from the axis or pinion that carries the minute-hand and therefore makes one revolution in an hour. Thus if a watch is required to go for 30 hours, the barrel must cause the centre pinion to rotate 30 times while the mainspring runs down; or, rather, during the four revolutions that are allowed by the stop-work.

Assume the pinion to have 10 leaves; each leaf corresponds to a tooth of the barrel and thus 300 teeth must pass in the 30 hours.

Now if the stop-work allows of four complete rotations of the key, it is clear that these 300 teeth must be given by four revolutions of the barrel; that is to say the barrel must have $\frac{300}{4}$ or 75 teeth.

Keeping the same barrel and stop-work, if the 10-leaved pinion be replaced by one of 7, 8 or 9, the period of going will be gradually increased, and, conversely, it will be diminished if pinions with 11, 12, etc., leaves be employed. This is evident.

1062.—To recapitulate then; in order to ascertain the number of hours that a watch will go, *multiply the number of teeth of the barrel by the number of turns allowed by the stop-work* (or the turns of the spring if no stop-work exists) *and divide the product by the number of leaves of the centre pinion;* the quotient gives the required period in hours.

Example.—A barrel has 80 teeth and its stop-work gives 4 turns; the centre pinion has 12 leaves.

$80 \times 4 = 320$; this number divided by 12 gives 26·6, that is to say 26·6 hours, or 26 hours 36 minutes.

1063.—*Conversely.*—If it be required to determine the number of teeth of the barrel in order that the watch may go for a certain number of hours, *multiply the number of hours by the number of leaves of the pinion, and divide by the number of turns allowed by the stop work.*

Example.—Let a watch be required to go for 30 hours with a centre pinion of 8 leaves and a mainspring that makes four complete turns: the number of hours (30) multiplied by the number of leaves (8) gives 240, and this product divided by 4 gives a quotient, 60; this is the required number of teeth of the barrel.

1064.—*For a watch with a fusee* proceed as if the fusee were the going barrel, and ascertain the number of turns of the fusee that correspond to the uncoiling of the chain from the barrel.

1065.—*To calculate the time of going of a clock.*—The calculation is similar but slightly more complicated, owing to the existence of an additional mobile.

Multiply the number of teeth of the barrel by the turns of the mainspring (or stop-work) and this product by the number of teeth in the time wheel, intermediate between the barrel and minute-wheel pinion. The resulting product is then divided by the product of the leaves of the two pinions, and the quotient gives the number of revolutions of the centre wheel. If this be divided by 24 we obtain the number of days that the clock will go, and there will generally be a fraction over, that should be converted into hours and minutes.

1066.—*For a Turret Clock*, multiply the teeth of the first wheel by the number of times the cord is wound round the drum, etc.

1067.—It does not enter into the province of this work to calculate complicated trains of wheels, such, for example, as those of orreries. Horologists that wish to occupy themselves with mechanisms of this class should have recourse to the works of A. Janvier* and to an excellent treatise published by M. Achille Brocot, entitled: *Calcul des rouages par approximation.*

CALCULATION OF THE FORCE TRANSMITTED BY A TRAIN OF WHEELS.

1068.—We will assume the train to remain in a statical condition, that is to say the power and the resistance (represented by two weights suspended from the circumference of the first and last wheel) maintain equilibrium. All the several lever arms are considered to be engaging on the lines of centres and

*Antide Janvier was born at Saint-Claude-du-Jura in 1751 and died in 1835. He was celebrated for his skill in representing planetary movements by the aid of mechanism. His singular skill was supplemented by profound mathematical knowledge.

the teeth to be of such a form that the conditions of the transmission of motion are the same as if the two pitch circles were rolling in contact (fig. 19, plate XIII.).

We will first give the principles on which the calculation is based.

1069.—In a train consisting of wheels and pinions, *the power is to the resistance as the product of the radii of the pinions is to the product of the radii of the wheels*, if we neglect friction.

1070.—The approximate *ratio of the power to the resistance, ignoring friction, may be found by measuring the space traversed by the two forces: these forces are to each other inversely as the spaces* (**27**).

1071.—Let A, B, C, D (fig. 19, plate XIII.), be a train of wheels. Take the half-millimetre as a unit of measurement o the diameters, and assume a weight of 160 grammes to be suspended to the circumference of A.

As this wheel, with a radius 20, has its two opposite lever arms of equal length, we may replace it by one of greater or less diameter, providing its weight remains the same; we will, then, assume the motive force to be applied tangentially to the circumference of the pinion p.*

The radius of B (16) is four times that of p (4). Thus the force transmitted by B is reduced by three quarters, that is to say it becomes 40 grammes. The radii of C (12) and D (10) are respectively four and five times the radii of their pinions, p' (3) and p'' (2). The force transmitted by C is, then, only a quarter of 40, or 10, and that transmitted by D is a fifth of 10, or 2.

If then a weight of 2 grammes be suspended from the circumference of D, it will balance a weight of 160 grammes at the circumference of A. The power is to the resistance : : 1 : 80.

1072.—By calculation we should have had,

Product of the radii of wheels $16 \times 12 \times 10 \parallel 1920$
" " " pinions $4 \times 3 \times 2 = 24$
Hence (**1069**) P : R : : 24 : 1920
And we have 24 : 1920 : : 1 : x $\therefore$ $x = 80$.

This shows that a force of 1 applied at the circumference of the last mobile, D, will neutralize a resistance or force of 80

* A wheel that receives and transmits force with arms of the same radius, which is often termed an *idle wheel*, does not enter into a calculation of the force transmitted since it only serves to alter its direction. It gives up all the force that it has received with the exception of the small amount needful to set it in motion.

applied tangentially to the first pinion, or, in other words, to the circumference of the wheel A; the result, then, is identical in the two cases.

1073.—On calculating the velocities of the wheels we see that D makes 160 revolutions while A makes 1. To determine the space traversed by a point on the circumference of A in the one revolution, we have (**137**), $40 \times 3{\cdot}14 \times 1 = 125{\cdot}6$; and for a point on the circumference of D during the 160 revolutions, $20 \times 3{\cdot}14 \times 160 = 10048$. But $125{\cdot}6 : 10048 :: 1 : 80$. The result, then, is the same as above (**1070**). (See the footnote on page 594.)

1074.—If the motor, instead of acting at the circumference of A, be applied at that of a pinion or pulley *s* on the axis of A, the discussion would be identical with that given above, except that the driving weight should vary in the inverse ratio of the two radii *r n*, *r s*. It will simply amount to taking *s* as the point at which the force is applied.

1075.—*Observations.*—In all the preceding calculations we have ignored the effect of friction; as a rule it absorbs in machines about a third of the driving power; but considerable divergence from this estimate must be looked for in horology. For, in consequence of the motive force being very slight, the resistance due to friction varies between very wide limits, as it depends on the nature of the materials employed, their hardness and degree of polish; the accurate adjustment of the depths, weight of the several mobiles, state of the oil, &c. If it be required to determine the amount of force that is dissipated with greater accuracy, it would be best to feel the pressure exerted at the extremity of a tooth of the escape-wheel by means of a very delicate lever, adjusted so as to balance this pressure.

1076.—By such means as these we can ascertain with a sufficient degree of accuracy the force exerted at the circumference of a wheel, in other words the statical pressure available at the extremity of an escape-wheel tooth.

The intensity of the action on the balance of this wheel when in motion, assuming it to occur uniformly, without impact or friction, would be given by multiplying the statical pressure by the square of its velocity. Hence it follows that the maximum effect will be obtained when the escape-wheel so acts on the balance as to reduce the resolution of force to a minimum, and to allow of the wheel acquiring, during the short period of

its motion, a velocity that is relatively sufficient (**28**). We cannot advantageously discuss this question further; its theoretical consideration requires all the resources of analysis, and, even if such a solution were obtained, it could only serve as a guide to an experimental method, because the instantaneousness of certain effects prevents our appreciating them, and the resistances due to friction, etc., in very small mechanism cannot, at the present day, be represented by any reliable figures.

1077.—From this brief discussion of the subject it will be seen that the designing of an escapement, a calliper, etc., not only requires a prolonged practical experience of horological mechanism, but also considerable theoretical knowledge; and we would take this opportunity of pointing out to those who occupy themselves with mechanisms that are complicated or intended to go for a long period, that, if their efforts are often unsuccessful, it is due to their inability to appreciate the increase in the motive force that is rendered necessary through the waste of energy in various ways, although the necessary calculation is, as we have seen, easily made; they, moreover, ignore the fact that adhesion, friction, stickiness of oil, especially between the coils of the mainspring, etc., further give rise to so many obstacles and to so much irregularity that *the angular movements become slower and the spaces traversed in a given time become less.*

CHAPTER I.

PRINCIPLES ON WHICH A DEPTH IS CONSTRUCTED. VARIOUS KINDS OF DEPTHS.

The forms of the teeth and leaves of mobiles.

1078.—Since a depth is, as we have already seen, nothing more than a system of pairs of levers succeeding each other without sensible intervals, it follows that this depth will be as nearly as possible perfect when:

The power arm and the resistance arm, acting at or very near to the line of centres, remain of the same relative lengths through the entire lead; neglecting friction, this would cause the amount of force transmitted to remain constant.

The velocities of motion are uniform and the pressure on the acting surfaces invariable;

And, lastly, the friction is reduced to a minimum and is disengaging.

1079.—We have already stated (**1045**) that if a wheel could drive another wheel by the mere contact or slight pressure of their circumferences (fig. 20, plate XIII.), we should have a perfect depth, because: (1) it would be accomplished by contact and without friction; (2) the radii of power and resistance being simply the radii of the two wheels, the ratio between these two forces would remain invariable; (3) the power would always act exactly in the opposite direction to the resistance; (4) the velocities would be uniform, since each portion of the circumference of the driving wheel would carry forward an equal portion of that of the other wheel; and (5) the pressure would remain constant.

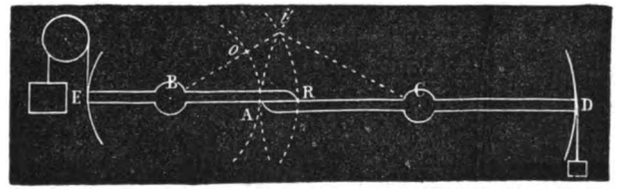

Fig. 65.

1080.—It has been shown in article **1046** why this method of transmitting force cannot be made available in practice; and the necessity for providing the circumference with teeth was also explained.

Let us now consider two levers, having their fulcra or axes of rotation at the points B and C (fig. 65), the middle points of the lines E A, A D, and let the extremities D and E be circular arcs concentric with these axes of rotation; it will be seen that this system accurately represents the case of two teeth of a depth where the teeth are left square, thus corresponding to straight levers.

We would observe, before proceeding further, that if any movement be imparted to the levers, it will not in any way modify the length of the arms C D, B E, but these will always remain the same, each being the radius of its own circle. We have, then, only to consider the two arms A C and B R.

If a movement be communicated to the longer arm, A C, so as to carry the point A to *i*, the short arm, B R, will move into the position B *i*. In the first instance the acting arms are A C and A B; at the termination of the movement above indicated one of them, A C, has retained its initial length, whereas the other has been increased by the quantity *o i*. Evidently this difference in the length of the arms must occasion a corresponding change in the relation of the power to the resistance, and the initial equilibrium will thus be disturbed.

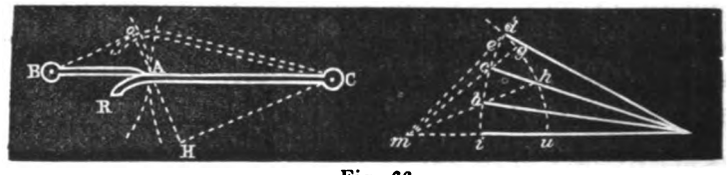

Fig. 66.

The velocities, moreover, change; for, if the long arm be assumed to move with a uniform velocity, the movement of the short arm becomes gradually slower, as will be easily seen from an examination of fig. 66, where the equal circular arcs *i b*, *b c*, *c d*, etc., traversed by the extremity of the long lever arm, correspond to the gradually diminishing angles *u m h*, *h m g*, *g m d*, etc., traversed in equal intervals of time by the short arm.

1081.—Whenever a straight lever acts on another straight lever these inconveniences are experienced; but such is not the case if, for one of the straight levers, we substitute a bent lever of such a form that, when the curved portion A R (fig. 66) impels the short arm B A, we always find (at whatever position they be considered): (1) that the short (virtual) arm B *o* (measuring from B to the point of contact, *o*) is to the long (virtual) arm H C,

as B A is to A C; and (2) that the arc A *o*, traversed by the arm B A, is equal to the arc described by the point A of the long arm. The problem would then be completely solved, for the power and resistance would always remain in the same proportion, as would also the velocities.

Among the several curves that are applicable to depths, four possess special interest to the watchmaker: these have been explained in articles **1037**, **1040** and **1041**, and they are employed in forming teeth that are termed helical, involute, cycloidal and epicycloidal.

HELICAL TEETH.

1082.—Helical teeth, when formed in accordance with the conditions laid down by James White, act on the line of centres or at least infinitely near to it, without impacts and by the mere superposition of the points of contact, in other words by pressure and rolling without any sliding friction. They satisfy all the demands of theory; but experience shows that, when the pressure is considerable, the acting surfaces, being of small extent, scratch each other; whereas, at the last mobiles of a horological train, where the pressure is slight, any dust or foreign matter settling on the surfaces will remain there, in consequence of the action consisting in a mere superposition; ultimately forming a layer that gives rise to adhesion between these surfaces.

An ordinary depth is not subject to such an inconvenience; for, however slight its friction may be, it will suffice for a long time to force particles of dust, etc., away from the surfaces of contact.

Although helical teeth have not satisfied the expectations of some watchmakers, they may nevertheless be employed with advantage when the movement of a train of wheels is required to take place uniformly, without impacts and with a minimum of noise (Chapter VI.).

INVOLUTE TEETH.

1083.—The depth in which the points of the teeth are involutes of circles transmits the motion with a constant pressure and without any modification or loss of power; a fact that is due to the property of this curve (already indicated in article **1040**) in virtue of which the two virtual radii of power and resistance are always equal to the primitive or geometrical radii.

This remarkable property would have rendered the involute preferable to epicycloidal and hypocycloidal teeth if it had been possible to form the teeth and leaves accurately in accordance with the required curve. By employing an apparatus termed the *Arrondisseur de précision*, an improved rounding-up tool which we have invented, such teeth might be made, but it would be necessary also to polish the leaves of pinions on the tool; and such a mode of proceeding would be impracticable so long as watches are produced in large quantities as at the present day.

When a wheel is called upon to drive several pinions, involute teeth should be adopted, since the forms of epicycloidal teeth vary in accordance with the size of pinion.

Involute teeth may be advantageously employed in turret clocks and especially in remontoirs. In the smaller kinds of horological mechanism, where they could only be adopted with difficulty, if the mobiles have a sufficient number of teeth of the epicycloidal form they will very closely resemble involute teeth (**1123**). (See chapter VI.)

EPICYCLOIDAL TEETH.

Selection of the most convenient curves.

1084.—On examining fig. 6 (plate XIII.) where the large circle, A, represents a wheel and the small circle, B, a pinion, we see that, if the large circle be made to rotate, it will cause the two smaller circles, B and C, to revolve at the same time. The movement will be the same as if the smaller circles roll round the larger circle while it is maintained at rest.

It is manifest that throughout the movement the small circle, C, will roll not only round the great circle, A, but also round the interior of the circle B, and the generating point therefore will at the same time trace out the epicycloid *a o b* and the hypocycloid *c o d*. But these two curves, since they are generated by one and the same point, must necessarily be always in contact at that point and that point only, and therefore the line joining *m*, where the three circles touch, and the point of contact, *o*, of the epicycloid and hypocycloid, will always be a normal to these curves.

1085.—If the diameter of the small circle, C, be greater than half the diameter of B, the hypocycloid *c o d* will be convex (fig. 6, plate XIII.)

If less than half that of B (fig. 5, plate XIII.) the hypocycloid *c o m* is concave.

And if the diameter of the small circle be half that of B (fig. 4, plate XIII.) the curve becomes *a straight line, i d*, and this line is a diameter of the circle B.

The latter very curious case materially simplifies the theory of depths and facilitates its application.

In short, in order that the depth of the wheel A and pinion B may possess the advantages indicated by theory, it is necessary that the side of the pinion leaf be hypocycloidal, and that of the wheel tooth be epicycloidal and projecting beyond the circumference A.

But with the circle C (fig. 6) the side of the pinion leaf would be projecting; with G (fig. 5) it would be hollowed out, but in the last case, with the circle H (fig. 4), the side would become a straight plane directed towards the centre of the pinion. But, although it may be difficult or even impossible to form the sides of leaves in accordance with any predetermined curves, such is not the case with a straight plane, which can be produced even on a small scale without the least difficulty.

Those of our readers that have difficulty in understanding the above explanations should have recourse to the practical method already indicated (**1038**).

1086.—*Résumé.*—With epicycloidal teeth it is essential that:

(1) The curved points of the teeth be traced out by a circle whose diameter is half the pitch circle of the pinion;

(2) The sides of the pinion leaves be straight planes directed towards the centre of the pinion.

In articles **1104-5** will be found the different methods adopted for tracing the epicycloid.

Advantages and disadvantages of Epicycloidal Teeth.

1087.—If the faces of the leaves be made hypocycloidal and of the teeth epicycloidal, the depth will be characterized by the following features:

(1) The line *s o e* (fig. 4, plate XIII.) being perpendicular to both curves, the force of the wheel will always be applied perpendicular to the leaf of the pinion.

(2) The velocities are equal, since the length of the arc *i o*, through which the pinion has revolved, is equal to that of *r o*, through which the wheel turned in the same interval of time.

(3) Throughout the whole period that the tooth and leaf are in contact, in other words during the entire lead, the power

and resistance maintain a constant proportion. This may be proved as follows: from the principle of the lever (**1034**) it follows that the lever arm by which the wheel acts is not its exact radius, but the virtual arm, v *s* (fig. 4, plate XIII.); this must, therefore, be regarded as the power lever; and the resistance lever, instead of being the radius of the pinion, is only that portion of it that is comprised between the centre *d* and the point *e* where the leaf and tooth are in contact. It can be geometrically shown that, at whatever point of the lead we arrest the tooth, the virtual arm v *s* is to the virtual arm *d e* as v *o*, the radius of the wheel, is to *d o*, the radius of the pinion: hence it necessarily follows that the power and resistance remain the same throughout the lead.*

1088.—These several advantages are accompanied by a slight amount of sliding friction, as will be evident from the fact that *r e* is of greater extent than *i e*. But this sliding, which becomes less and less as we increase the number of teeth (with less than thirty teeth a wheel can never drive a pinion uniformly), must be considered as an advantage rather than a disadvantage in the smaller class of horological mechanism (**1082**); the most serious inconvenience consists in the difference of pressure at the beginning and end of the lead, a difference which, after a long period, gives rise to the surfaces of contact being more worn at one part than at another (**38**).

1089.—The epicycloidal tooth has then very important advantages; it is the best that can be employed in small mechanism because it is the only one that we have hitherto been able to make with accuracy; at the same time certain horological authors are wrong in asserting that it secures a *perfect depth;* this is only approximated to when the mobiles are *high numbered*, and inconveniences arise when we exceed certain limits in this respect (**1124**).

Depths in which the lead commences before the line of centres.

1090.—The line of centres is, as we have already seen, a line assumed to pass through the two centres of rotation, that is through the centre of the wheel and pinion (B A, fig. 2, plate XIV.).

* The two right-angled triangles *o* v *s*, *o d e* (fig. 4), are *similar;* hence their two homologous sides are proportional, and we always have *s* v : *e d* : : v *o* (the radius of the wheel) : *d o* (the radius of the pinion).

If it be required that the impulse applied to the resistance arm be only accompanied by disengaging friction, the tooth must not come in contact with the leaf until the face of the latter is on the line of centres (as X *o* touches Y *o* in fig. 3).

This will always be the case with pinions of 12 leaves or more, and even with only 10 or 11 leaves providing they are formed as we shall presently explain; but when the pinions are low numbered, 9, 8, 7, 6, the lead will always commence before the line of centres by an amount that increases as the number is reduced. This may be proved as follows:

Let A (fig. 17, plate XIII.) be a wheel of 40 teeth; each tooth and adjacent space (*a m* or *a n*) will occupy on the primitive circle an arc of 9° (for $9 \times 40 = 360$, or the number of degrees in a circle).

On examining this figure 17, in which the length of the arc *a n* is the same as that of *a o* with an 8-leaved pinion, it will be evident that, if the teeth and spaces be made of equal width, the tooth B will not impel the leaf F far enough, but the following leaf H will engage with the tooth C at some distance before the line of centres. If we increase the width of the tooth B so as to obtain a longer ogive, the new tooth *n r i* will be much wider than the space *i a*, and its point *r* will drive F far enough to prevent C engaging with the face *p a* before the line of centres; but in that case it will be necessary to reduce the thickness of the leaf by the same amount as the tooth is increased, for, otherwise, the tooth would butt against the corner of the leaf and movement would become impossible. The leaf, then, must be made very thin.

1091.—For the lead to commence on the line of centres with an 8-leaved pinion, about 5/6^{ths} of its circumference must be spaces; in other words, the width of a space must be five times that of a leaf.

With a 7-leaved pinion, the leaf must be still thinner.

With a 6-leaved pinion, it would become absolutely nothing.

From the preceding considerations it will be evident that with low numbered pinions the lead will always commence before the line of centres; for it is of the first importance to retain the leaves of a certain thickness, both on account of solidity and in order to avoid distortion in hardening. It is a recognized rule in practice that a low numbered pinion cannot have more than 2/3^{rds} space (measured along the arc *a o*, fig. 17), and thus that the thickness of the pinion be one-third.

It naturally follows from the above considerations that the lead will commence nearer to the line of centres as the leaf is made thinner, the tooth thicker and therefore its point higher.

1092.—Since a low numbered pinion cannot secure a depth free from engaging friction, it is important when the transmission of force is required to be very uniform, etc., only to employ pinions of 10 leaves or more, because with such pinions the lead is after the line of centres. At the same time, although low numbered pinions must be carefully excluded from all the very best work for this reason, they can be employed in ordinary watches. If they are made in accordance with the rules given subsequently, well hardened and polished, they give very satisfactory results; but it must even then be remembered that one of 8 is preferable to one of 7 leaves and one of 7 gives better results than one of 6 leaves.

Reason for rounding the point of a pinion.—On the best height for the ogive of a tooth.

1093.—It will be remembered that by the *point* is understood the entire portion of either a tooth or leaf that projects beyond the pitch circle; *r a o* and *i z n*, fig. 4, plate XIII.

The point of a wheel tooth, which is also at times known as the *ogive* on account of its form, must, as we have already seen, be of a certain epicycloidal form; the point of a pinion leaf, known as the *rounding*, has not yet been discussed; if it be desired to make pinions of low numbers strictly of the theoretical form, the points must also have an appropriate epicycloidal form. But it should be remarked that: (1) in pinions of moderately high numbers, as we shall subsequently show, the point only serves as a precaution; (2) it is almost impossible to form the point of the pinion leaf in horological mechanism strictly in accordance with the requirements of theory; (3) an infinitely small portion of the point is touched by the tooth as it enters a space.

These facts have led makers to adopt the semicircle as the most convenient form to give to the point of the leaf; it is then simply rounded off as shown in figs. 1, 2, 3, 4, & 6 of plate XIV.

1094.—Rounded pinion leaves have several advantages when employed in conjunction with epicycloidal teeth: (1) they facilitate the accurate determination of the difference between the

total and primitive diameters, since the one differs from the other by the thickness of a leaf; (2) it is possible to pitch low numbered pinions sufficiently deep without any serious increase of friction; (3) it not only is easier of construction but also indicates accurately the positions of the pitch circles, which must be known in order to secure a good depth.

1095.—When the pinion is high numbered, the ogive of a tooth will never lead the leaf with its point, because another tooth engages with the succeeding leaf before the first tooth has accomplished its lead; hence it follows that the contact of a leaf and tooth is of shorter duration as the mobiles are more highly numbered, and that, in depths of this description, a portion of the points of teeth is useless and may be suppressed (fig. 2, plate XV.).

1096.—With pinions of 10 leaves and under it is essential that the point of the ogive be retained; with 11 or 12 leaves it is barely possible to lightly remove the edge from the points of the teeth, so that it is generally retained as a precaution. Rather more can be removed with higher numbers, the amount being increased as the number of leaves becomes greater.

1097.—An observation contained in article **1084** affords us a means of ascertaining the best length for the ogive.

We know from previous considerations that the point of contact of the tooth and leaf corresponds with the generating point of the curve; draw the depth then (in the manner subsequently indicated) so that the face of the tooth *o* X (fig. 3, plate XIV.) touches the leaf Y on the line of centres, A B; then describe the generating circle D, and the point at which this circle cuts the face *e* of the leaf E determines the extreme effective point of the lead by the tooth G; for the tooth X *o* is then commencing its lead and the entire portion beyond *e* may be suppressed, as we have indicated by the circular arc P Q, described from the centre of the wheel.

The line *o e* will be perpendicular to the face *e* of the leaf E.

On the freedom at depths.

1098.—In a theoretical depth there is no freedom whatever between the teeth and leaves. For mathematicians calculate the dimensions and mode of action of the depth with absolute accuracy, and assume the construction to be as rigorously correct as the calculation, which, as we know, cannot be the

case. Moreover the continual change in matter through heat, the imperfect nature of our tools which does not permit of our obtaining perfect accuracy of form, and the friction of both sides of the tooth, would inevitably prevent the working of a depth without freedom.

If the teeth and leaves are made of the proportions subsequently given, the freedom will be found to be sufficient; for its actual amount may be assumed to depend on the inequalities in the subdivision of the circumference. In conclusion, it is only by practice and a systematic examination of well-made teeth that this amount can be ascertained with exactness.

An Observation of Camus.

1099.—"As we can never hope to form the teeth with such accuracy that the pitch circles of the wheel and pinion shall always rotate with equal velocities, and as the inequalities and other faults in the teeth will make the lead measured from the line of centres in some cases not sufficiently long, so as to occasion buttings, etc., makers will do well to avoid these inconveniences by making the primitive diameter of the mobile that drives slightly greater than it should be in comparison with the mobile that is driven.

"By this increase in the diameter of the wheel, which should be proportioned to the faults that are expected in the form of the teeth, the tooth immediately succeeding that which is leading a leaf beyond the line of centres will engage with the next leaf somewhat later and, when the first tooth has driven the pinion as far as it can uniformly, the wheel has a somewhat greater velocity than the pinion, and this is a fault; but this intentional fault is less objectionable than the buttings that would probably occur if it were not to exist."

CHAPTER II.

TO DESIGN A DEPTH.

1100.—It is now easy, with the help of the theoretical and practical data at which we have arrived, to determine the most suitable proportions for a depth and to draw it.

Watchmakers, especially the younger members of the trade, cannot be too much practised in the correct drawing of depths; there are few methods more efficacious for bringing home to them both their principles and mode of action.

It is first necessary to measure the distance between the centres of the two mobiles on the calliper of the watch, etc., and to fix upon the number of teeth or leaves for each, numbers which are determined by the revolutions of the pinion that correspond to one revolution of the wheel.

Let the pinion be required to rotate eight times during one rotation of the wheel, so that the wheel has eight times as many teeth as the pinion has leaves;—48 teeth for a pinion of 6 leaves; —64 teeth for one of 8 leaves;—80 for one of ten leaves, and so on.

Taking the numbers 6 and 48 and the distance between the centres 15 millimetres, draw on a sheet of stout well-stretched drawing paper a straight line A B (fig. 2, plate XIV.), on which mark the two centres A and B, 15 centimetres apart. (The drawing will then be magnified 10 times and it will only be necessary to divide its several dimensions by this number to obtain the actual proportions of the depth. It would be still better to magnify the drawing 20 or 30 times.)

As shown in article **1049**, we obtain the proportion $48 + 6 : 15 : : 48 : x$, or $54 : 15 : : 48 : x$. Hence we have $x = \frac{15 \times 48}{54} = \frac{720}{54} = 13{\cdot}33$.

The primitive radius of the wheel will then be 13·33 centimetres, and deducting this amount from 15 centimetres, or the distance between the centres, we have 1·67 centimetres as the primitive radius of the pinion.

With a radius slightly in excess of 13·33 centimetres (**1099**), say 13·5 centimetres, and from the centre A describe the circular arc *c a f;* then from the centre B describe the second circular arc *b a e*, touching the first on the line of centres.

These will be the pitch circles of the wheel and pinion respectively.

Now since the pinion has 6 leaves and of necessity the same number of spaces, a leaf and an adjacent space should occupy one-sixth of the 360° in the circumference of the pitch circle, that is to say 60°; and, as we shall presently show that the spaces in a pinion of 6 leaves should be twice the width of the leaves, it follows that the leaf will measure 20°. As a leaf must be touched 10° before the line of centres, draw the line B *a* making an angle of 10° with the line of centres B A; then draw B *n* making an angle of 20° with B *a* and thus determining the thickness of the leaf.

Angles *b* B *a*, *n* B *e*, of 40° each on either side of this leaf will mark off the spaces; and then the faces of the two leaves, H and L, can be drawn, etc.

The point of each leaf will be obtained by describing a semicircle, taking as a centre the middle point of the portion of the pitch circle intercepted between the two faces of the leaf.

The depth of the spaces cannot yet be ascertained because it depends on the height of the ogives of the wheel teeth, and this we have not determined.

As the point *a* on the leaf M is the first touched, the straight line *a* A drawn from *a* to the centre of the wheel will give the face of a tooth.

Now this wheel has 48 teeth, so that a tooth with the adjacent space will extend over $\frac{1}{48}$th part of the circumference, that is $\frac{360}{48}$ths or 7·5 degrees. Having divided off the arcs *c a*, *a d*, *d f*, of 7·5° each, bisect them by the lines *p*, *r*, *s*, if the teeth and spaces are required to be equal, and with a rule passing through the centre A and the several points *c*, *p*, *a*, *r*, *d*, *s*, *f*, draw the faces of the wheel teeth.

We now require to fill in the ogives.

On a separate piece of cardboard (fig. 1, plate XV.) draw the arc A *g* C B with the primitive radius of the wheel. Commencing at the extreme point of the radius E *g* form the required epicycloid *g d* (**1104**); then cut out the cardboard along the line E *g d* and apply it to fig. 2, plate XIV. so that the line E *g* coincides with the line A *p* of fig. 2, and the arc A *g* C B with the pitch circle of the wheel: draw in one side of the ogive, and it will only be necessary to reverse the pattern or templet in order to form the other side in the same manner.*

* The drawing being supposed to be prepared with the utmost accuracy, more especially as regards the division of the circumferences, it may be verified as follows.

From the centre A, fig. 2, describe the dotted circle x z passing through the points of the ogives. This circle, since it determines the pitching of the teeth with the leaves of the pinion, will enable us to fix upon the depth of the pinion spaces. A circle described from the centre B and passing through the extreme points of the leaves (P V K) will give the depth of the wheel spaces. Of course a sufficient amount of freedom must be left to ensure safety between the bottom of the spaces and the extremities of the teeth and leaves.

The depth has now been completely drawn, and, if the operation has been performed with care, it will only be necessary to take its exact dimensions (the pitch circles, total diameters, thickness of leaf, height of ogive, etc.) and to reduce them by the aid of a proportional compass or by calculation, and we shall possess the dimensions of an excellent depth satisfying the given conditions (chapter VI.).

Observations.

1101.—As was the case with escapements, the drawing should be first made with a fine pointed pencil, in order that any errors may be corrected. It is unnecessary to mention that only those lines which are required to remain should be drawn in with Indian ink.

At the centres of the wheel and pinion a small flat disc of some hard substance on which the centre is marked by a point, mother of pearl for example, should be fixed in the first instance with mouth glue in order to avoid enlargement of the centre or displacing it by the point of the compass. Or the thin metallic centres used by draughtsmen may be employed; they are held in position by three very fine points.

The height of the ogive (chapter IV.) or the line *c e d f* (fig. 13, plate XIII.) being known, and the straight faces of the

If the tooth *r d* touches the face *e t* with its point, the height of the ogive is correct; but if the point *e* is at a distance from the face *e t*, the ogives are too short and the lead will commence before the point *a*.

If the point of the tooth overlaps the surface of the leaf, the ogives are too long.

When a line *j t* is drawn perpendicular to the face *t e* from the point at which the line of centres crosses the pitch circles (**1097**), we may (in the case of pinions under 10 leaves) take the distance between *t* and the point of the tooth as practically equal to the distance of *a* from the line of centres. With a pinion of 10 leaves the point of the tooth will be at *t*, and with one of 12 leaves it will be somewhat above this point.

teeth having been drawn in, as well as radii passing through the middle points of the teeth, it is a common practice to replace the epicycloid by a circular arc embracing two teeth when the accompanying pinion has less than 10 leaves, and three teeth if from 10 to 18 leaves.

The following is the method of procedure:

After drawing the straight faces of the teeth, as well as the central lines *c g*, *e b*, *d i*, *f j* (fig. 13, plate XIII.), set the compass so that with a centre somewhat within the pitch-circle (*x* or *x'*) a circular arc may be described passing through the points of intersection *c*, *e*, and falling just beyond the faces *n*, *r*. Then make the straight and curved lines fit together properly and the curve thus obtained will differ very slightly from a true epicycloid (**1149**).

With high numbered wheels, where a portion of the ogive is suppressed, the amount to be removed should be indicated (fig. 2, plate XV., **1097**), because it will enable the spaces of the pinion to be made less deep and thus its solidity will be increased.

A single drawing of a depth will suffice for all depths of the same numbers whether the teeth are large or small. Thus if the distance of the centres is 30 instead of 15 it will only be necessary to double the dimensions, and, conversely, they must be reduced one half for a centre distance of 7·5 . . . etc. This is an operation of no difficulty as will be subsequently seen.

1102.—We have just given general rules for the designing of depths, but it should be pointed out that the process is still further simplified by following the directions and adopting the proportions, etc., given in chapter IV.

We would here observe to those watchmakers, unfortunately very numerous, that consider a large scale drawing of a depth to be useless, pretending that it is impossible to accurately reduce the proportions and apply them to the depths of a watch, that they are entirely mistaken. The reduction can be made with ease by calculation and carefully graduated gauges, etc., such as are described farther on. As regards the carrying out of the principles on a small scale, it requires very great care but presents no difficulties, since the watchmaker has at his disposal means sufficient to satisfy the requirements of chronometer-making and therefore of the more ordinary horological mechanism.

Another mode of ascertaining the relative lengths of the primitive radii.

1103.—Knowing the distance between the centres we can, instead of employing a proportion to determine the length of each primitive radius, as was done in articles **1049**, **1100**, divide this distance into one more part than the number obtained by dividing the leaves of the pinion into the teeth of the wheel. Then take one of the parts so divided for the primitive radius of the pinion and the remainder for that of the wheel.

Take the example of article **1100**.

The numbers of the wheel and pinion are respectively 48 and 6, so that the latter is contained in the former 8 times: the distance between the centres (15 millimetres) must then be divided into 9 (8 + 1) parts. Dividing 15 by 9 we obtain 1·67 millimetres very approximately, and this is the radius of the pinion.

The difference between the result here given and that obtained in article **1100** is less than a half-hundredth of a millimetre, an amount which is quite insignificant, and it can be still further reduced by carrying out the division to a greater number of decimals.

To draw an epicycloid by means of successive points.

1104.—Let A C B (fig. 1, plate XV.) be the pitch circle of the wheel, and D the generating circle. Draw a line E *g* O passing through the centre of the generating circle and the first point of contact. Divide an arc *g* C (somewhat greater than the width of the tooth) into equal parts 1, 2, 3, 4, etc. by the aid of a compass. Now draw through these points and the centre E of the wheel lines 1 *m*, 2 *n*, 3 *p*, etc.; and from E describe the circular arc O R passing through the centre of the generating circle. Each point of intersection of the arc O R and the lines 1, 2, 3, etc. becomes the centre of a new position of the generating circle, and these should be traced out.

Accurately measure with a compass the distance *g* 1, and, taking 1 as a centre, mark it off on the generating circle 1′. Then with the centre 2 mark off twice the distance *g* 1, that is *g* 2, on the circumference 2′. Do the same on circumference 3′ and so on.

The curve *g d* passing through the several points so obtained will be the required epicycloid.

REMARKS. The parts into which *g* C is divided should be as small as possible, in other words no larger than is necessary to ensure accurate drawing. Short arcs can, without sensible error, be regarded as portions of straight lines.

The compass should be light and its points very fine. It must not be raised from the paper for each measure but should rotate on one of its points, always maintaining the eyeglass to the eye. Further, it may perhaps be well to add that in marking off a point, say 10′, the 10 short arcs equal to *g* 1 must be marked off in succession *along the arc* C *d*, and not by a single measurement of *g* C.

To trace out au epicycloid by rolling.

1105.—The circle A (fig. 4, plate XV.) is part of the pitch circle of a wheel cut in wood, of which the circumference is perfectly true. A second wooden disc is turned on the lathe, having the diameter of the generating circle. A thin ribbon *c d*, that is not liable to stretch, is fixed by one extremity to the wheel and by the other extremity to the generating circle. A blunted brass point, *o*, is in the plane of B, so arranged that it can project slightly below the disc B at its edge.

The system thus arranged is laid on a stretched sheet of paper (preferably covered with oxide of zinc on which brass leaves a black trace) and the disc A should be firmly fixed on it. Then connect the two axles M and N (fig. 3) by the cord *d h* in order to prevent the two discs from separating.

After stretching the ribbon very tight and placing the point *o* on the line of centres (fig. 3), rotate the circle B towards the right pressing gently on its face. The point *o* will trace out an epicycloid on the paper with sufficient clearness if the arrangement has been carefully and intelligently made. It will of course be evident that the ribbon should be as thin as possible, because when rolled round the disc its thickness is added to the radius of this disc.

To trace out a cycloid.

1106.—The cycloid may be described in precisely the same manner except that the circle A is replaced by a straight rule.

CHAPTER III.

TABLE OF THE SIZES OF PINIONS; VARIOUS DETAILS CONCERNING THE LEAD, THE DROP, AND HIGH NUMBERED WHEELS.

Sizes of pinions (ordinary method).

1107.—We give here the table usually employed for ascertaining the sizes of pinions, subject to the following reservations. They are sufficiently accurate to secure us against the most serious cases of stoppage; and this is all that can be reasonably hoped from workmen who are too badly paid to justify us in demanding either more work or better quality; but we would here point out to manufacturers that the two following chapters will afford them ample means of ascertaining the true size, in other words, the size that secures the most perfect depth.

1108.—To give the diameter of a pinion approximately the pinion calliper should include:

For 16 leaves, 6 full teeth, that is to say measuring the distance between the two external faces;
„ 15 „ rather less than 6 teeth, or 5 teeth and just beyond the point of the sixth;
„ 14 „ 6 teeth measuring at the points;
„ 12 „ 5 teeth measuring at the points (or rather $4\frac{1}{2}$ teeth);
For a clock wheel 5 full teeth;
„ 10 „ 4 full teeth;
For a clock wheel, 4 *squared* teeth;
„ 9 „ rather less than 4 full teeth, or 3 full teeth to the point of the fourth;
„ 8 „ 4 teeth measured at the points minus a quarter of a space;
„ 7 „ rather less than 3 full teeth;
For a clock, 3 full teeth plus a quarter of a space;
„ 6 „ 3 teeth measured at the points or rather more.
For a clock, 3 full teeth.

1109.—Jurgensen after giving this table, which he took in great part from Berthoud, adds: "When the pinions drive the wheels they should be somewhat larger." This vague information only serves to lead into error the great majority of watchmakers that act on it to the letter. We shall show in chapter

VI. in what manner the size of a pinion that leads differs from that of one which is led.

1110.—In giving this table Berthoud was so far conscious of the insufficiency of the method that he added: Before hardening the pinion the depth must be examined on the depthing tool in order to accurately fix upon the size of the pinion and to give the best curvature to the points of the leaves; which only goes to prove that he arrived at the primitive diameter by frequent trial.

Imperfectness of the means usually employed for ascertaining sizes of pinions.

1111.—In the ordinary practice of horology, it is customary, as we have seen, to take as a measure of the total diameter of a pinion an opening of the pinion compass that covers on the circumference of the wheel a number of teeth and spaces equal to one-third the total number of leaves and spaces of the pinion.

Thus in a pinion of 6 leaves there are a total of 12 leaves and spaces and a third of this number is 4. The compass must therefore cover 2 teeth and 2 spaces, corresponding to the amount ordinarily taken, 3 teeth measuring at the points, for there are then included between the points of the compass: two spaces, one full tooth and two half teeth, or two teeth and two spaces.

1112.—This method of measurement was based on the following facts:

"The circular arc on the pitch circle of the wheel that includes a tooth and space, that is to say the pitch, is equal to the pitch of the pinion; and since the diameter is one-third of the corresponding circumference, if we measure on this circumference of the wheel a distance equal to one-third the number of leaves and spaces of the pinion, we shall have the diameter of this latter."

1113.—This method is inexact, for:

(1) A diameter is not accurately one-third of its corresponding circumference; (2) the measurement taken is not that of an arc but of the chord of that arc; moreover, if we assume the chords to be equal, it by no means follows that the arcs would be equal, for those of the pinion are more curved than the arcs of the wheel pitch circle; (3) the ratio between the two geometrical diameters is not the same as that of the two total diameters, and the measurements are taken at one time on the

pitch circle of the wheel and at another time on the external circumference; (4) the actual total diameters vary with the thickness of the leaves and teeth since the height of the point depends on this thickness while the pitch circles remain invariable, etc.

These facts lead us to the following conclusion: the actual measurements in vogue give a 6-leaved pinion that is too small, and in order to avoid the inconveniences that might arise from this fact the leaf has to be rounded off at its end to rather more than a semicircle. The exact amount of this rounding is thus left indeterminate, and the solution of this very important point becomes arbitrary, dependent on the degree of intelligence of the workman.*

Some of the practical measurements given in the above table are rather less inaccurate as regards the total dimensions than those relating to a 6-leaved pinion; and we would further observe that authors vary among themselves, not always giving the same measurements for the same pinion.

Origin of the difference in measurement for clocks and watches.

1114.—The fact of authors recommending that the pinions of clocks should be left slightly larger than those of watches has encouraged the opinion that theory admits different principles in watches and clocks, and that, therefore, the laws of theory are arbitrary and uncertain.

The pinions of watches, clocks and timepieces must always be made *in accordance with the unvarying principle that the primitive diameters are in proportion to the number of teeth*, and the other dimensions, such as the thickness of leaves, height of points, etc., as laid down in the precise rules already in part given, which will be made more complete subsequently.

1115.—The difference in this matter originated as follows; it was erroneously explained by Berthoud:

(1) As we increase the number of teeth of a wheel, the

* Since the measurements taken in practice only give the total diameters, it follows that if three pinions are intended for the same wheel but one has thin leaves, another thick leaves and the third barley-shaped leaves, they may nevertheless have the same total diameters but their primitive diameters will differ, so that the rounding will descend lower in one case than in another; the thick leaf will be struck farther from or nearer to the centre than the thin leaf, etc. It will inevitably result that, if one of the pinions gives a good depth, the two other pinions will be very bad, notwithstanding that all three would be strictly in accordance with the so-called rule (**1122**).

difference between its total and primitive diameters becomes less, since the point or ogive is shorter as the thickness of a tooth is decreased;

(2) If we assume that a space and two half teeth occupy the same space on the pitch circles of two wheels whose centres are at D and G (fig. 9, plate XIV.), we see that, if the points of the ogives in the large wheel are at *a* and *b*, those of the small wheel will be at *s* and *c*, or farther apart. In other words, the circular arc intercepted between the points of the ogives becomes less and less (or approximates more and more to equality with *o n*) as we increase the diameter of the wheel.

It will be evident after reading these two observations that, if the diameters of pinions have been in the first instance accurately arranged for watch wheels which are small and low numbered, when we attempt to apply the same measurements to clock wheels, we must necessarily obtain a pinion that is too small, since these wheels are larger and of higher numbers than those of watches; and the difference has been found to increase as the measurements are taken nearer to the total circumference, that is the one passing through the points of the teeth.

Practical mode of ascertaining whether the lead is uniform.

1116.—Those who have any difficulty in understanding the causes of irregularity in the lead, can resort to a practical method of confirming the conclusions of science. Set in the depthing tool a wheel whose teeth differ as much as possible from those represented in the figures with a pinion that is either too large or too small. After having fitted on the centres of the tool two brass discs finely graduated at their circumferences (fig. 30, page 230), provide the axes of the wheel and pinion each with a finger that can rotate close to the graduated disc.

Now ascertain by employing an eyeglass the exact point at which a tooth engages with a leaf. After noting the graduation indicated by each finger, cause the tooth to perform about a quarter of its lead of the leaf, and again note the readings. Again set the wheel in motion through the same number of degrees as previously, and note the interval traversed during this period by the finger of the pinion. Each portion of the lead will be found to be represented by a different number of degrees on the wheel and pinion; and, if the pinion be made to travel through equal spaces, those traversed by the wheel will

no longer be uniform, so that we prove experimentally that each mobile moves with an irregular velocity, and that the depth cannot fail to be characterized by slippings, starts, etc.

As the wheels and pinions approximate to the correct theoretical proportions the depth will be smooth and free and will be characterized by a *uniformity in the lead.* This uniformity therefore constitutes one of the best means at our disposal for practically verifying a depth.

Lead without engaging friction when the pinions are low numbered.

1117.—Some horologists, being impressed with the disadvantages that result from a lead commencing before the line of centres, have suggested what they consider to be a method for making the lead always commence at this line. Many so-called watchmakers still seriously maintain that the one rule for depths is to make them engage *at the centre.* In addition to the absolute impossibility of making the lead commence at the line of centres when employing certain sizes of pinions, as we have already shown (**1091**), they only avoid one fault by falling into another that is greater. For in order that the lead should commence nearer the centre than theory makes it, in the case of low numbered pinions, it would be necessary to make the leaves of the pinions very thin, to make the pinions themselves very small, lengthen the teeth of the wheel beyond what theory indicates, and pitch the depth deeper. When these conditions are observed the lead will commence nearer to the line of centres; but the mobiles would at times move with different velocities since the lead is not uniform; there would, moreover, be more fear of catchings and the drop would be accompanied by slipping, especially towards the end of the lead. It is manifest that all these faults taken together would have a far more detrimental effect on the depth than a slight amount of engaging friction.

It will be well to adhere to the proportions given in the next chapter.

The drop; catchings due to pinions being too large or too small.

1118.—When the point of a tooth impels the leaf of a pinion, this point will slide rapidly along the face of the leaf,

so that the succeeding tooth falls with a sudden jump (the drop) on to the face of the corresponding leaf.

By the term *catching* is generally understood detrimental friction of the curved portion *o a* (fig. 15, plate XIII.) against F (the wheel turning towards the right), or of *e* against *d i* or of *e n* against F during the engaging action, or, further, of the point of a tooth against the bottom of a space. Some authors use the term to mean the same as engaging friction; with them therefore a catching is any friction whatever occurring before the line of centres (except butting action). This definition is not legitimate, for a catching always consists in a detrimental friction that must and always can be corrected, and it is well known that engaging friction cannot always be brought under this category. We include therefore as catchings: (1) friction of the back of a tooth during the engagement; (2) contact of the point of a tooth with the bottom of a space; (3) excessive engaging friction or friction occurring farther from the line of centres than theory requires; and these are all faults that should not be allowed to remain in a depth.

1119.—When the primitive diameter of a pinion is too small a drop will occur.

This can be easily explained; for if the pinion is too small the arc covered by a space and a leaf, or the pitch of the pinion, is less than the pitch of the wheel, and thus it happens that when the impelling wheel H (fig. 16, plate XIII.) is only pushing the leaf by the point of its ogive, the following tooth B is at a distance *a n*, from the corresponding leaf, a distance very nearly equal to the difference between the two arcs *a o* and *m n*. When in this position the pinion almost ceases to move, while the point of the tooth H, sliding along its face and escaping at the commencement of the rounding, causes the movement of the wheel to be accelerated until suddenly arrested by the engagement of the tooth B and the leaf A. The blow caused by this drop is all the greater according as the pinion is smaller and the mobiles travel more rapidly.

It will be observed that to diminish the effect of the drop it is only necessary to elongate slightly the straight face *o k*, for then the tooth H will not leave its leaf until the lead has commenced with the next tooth and leaf; so that the drop will be materially diminished or altogether done away with. It should be remarked that although a pinion may be of the proper

size, the lead will always terminate with a slipping action when the ogives are not sufficiently elongated.

1120.—When the primitive diameter of the pinion is too great, the pitch of the wheel (**1030**) will be less than that of the pinion. The velocities will differ and the tooth will engage with a leaf all the more in advance of the line of centres as the size of pinion is increased; this causes the engagement to be accompanied by catchings and, if the size of pinion is much too great, the catching may even resolve itself into a sort of butting action owing to the extremities of the teeth being forced against those of the leaves.

1121.—A butting will entirely stop the action of a depth.

A catching wastes a considerable amount of the motive force, causes rapid wear of the acting surfaces and often the machine is actually brought to rest.

Drops also cause a loss of force and rapid wear, but not to the same extent as in the previous case and they do not occasion a stoppage: hence we conclude, for the information of watch repairers, that if other things are equal in a depth drops will be less detrimental than catchings.

Whatever care be devoted to the application of theoretical rules in constructing depths, the imperfectness of our instruments and the errors caused by the hardening and polishing of pinions altering their form, always occasion some irregularities in a depth; and thus whenever a portion of a wheel comprising a space and tooth engages with a rather greater length of the pitch circle of a pinion, there will result a butting or catching action, and there will be a drop when the shorter interval is on the pinion.

We have seen above that moderate drop is less detrimental than catchings or buttings; it is therefore advisable to make the wheel a trifle larger than theory indicates to avoid these sources of error; the pinion being then somewhat small, the depth will have more freedom but will be characterized by slight drops (**1099**).

This suggestion must, however, be acted on with great caution; for it is impossible to give any rule to fix the amount by which the mobile that leads is increased, as it necessarily depends on the inequalities of division, and thus on the accuracy of the wheel-cutting and rounding-up tools, etc.

1122.—Practical Application.—If the preceding reasoning has been clearly understood it will be seen that a pinion is only

too large or too small when its primitive diameter is too large or too small. It is possible then to improve the character of the depth of a pinion that is small by filing away the leaf along a line such as B *o* (fig. 5, plate XIV.), so as to increase the primitive diameter.

It will of course be clearly understood that such a method is only here suggested for adoption by watch repairers on an emergency, and must not be resorted to in high class watchwork.

If the pinion is too large the rounding must be made deeper, giving it the form *a m* B *n d* (fig. 7, plate XIV.).

The *total* diameter of a pinion is, as will be evident, almost an arbitrary quantity, for it may be made a little larger or smaller providing that the curve at its point have a suitable form and that it terminates accurately at the commencement of the straight face, that is at the pitch circle.

Advantages and inconveniences of high numbered wheels and pinions.

1123.—It is well in accurate mechanism never to employ a pinion of less than 10 leaves because it is only above this number that a uniform lead can be secured unaccompanied by engaging friction.

As we increase the number of teeth of a wheel or pinion, the difference between the total and geometrical diameters becomes less because the teeth are narrower and thus their points less elevated.

A high numbered pinion will have a very short lead on each leaf. This lead measures 60° with a 6-leaved pinion, 45° with an 8-leaved pinion, 36° with a 10-leaved, 30° with a 12-leaved pinion, and so on.

It is evident that by employing mobiles that are more and more high numbered the lead becomes shorter and shorter, utilising a gradually diminishing amount of the curve of the ogive until it becomes very little more than a mere contact. Friction and even badly formed ogives may then be assumed to have very little effect.

As it is impossible in practice to give an absolutely exact form to teeth in watches, the best depths will be obtained by employing high numbered mobiles; but this rule must always be subordinate to the following facts.

1124.—There are limits to the rule we have just laid down;

for as the number of teeth is increased the amount of the freedom between the teeth and leaves just before they engage becomes less and the depth requires to be proportionately of more accurate construction. Every watchmaker will do well to remember this last observation, and he should further not forget that with high numbered pinions the least imperfection in the teeth may give rise to catchings.

This fact may be explained as follows:

(1) As the first point of engagement of the depth is brought gradually nearer to the line of centres, the tooth A and the leaf B (fig. 14, plate XIII.) will come nearer together when this contact occurs.

(2) When a low numbered pinion is replaced by one that is higher numbered, the total circumference is greater, just as *x c z* is greater than *m c n* (fig. 14). The extremity of the leaf thus comes nearer to the tooth A.

(3) As the number of leaves of the pinion is increased, the diameter of its wheel remaining the same, the generating circle becomes larger. It necessarily follows from this that the curvature of the points of the teeth becomes less and less marked.

CHAPTER IV.

PROPORTIONS OF PINIONS OF FROM 6 TO 14 LEAVES, WITH THE TEETH THAT ENGAGE WITH THEM.

Exact sizes of pinions.

Preliminary Observations.

1125.—When Camus published his *Traité des engrenages* the only escapements employed were characterized by a very marked recoil. With low numbered pinions the lead continues for the greater distance beyond the line of centres as it commences nearer to that line, and it follows that, towards the end of the lead, the points of the ogives are acting more and more perpendicular to the faces of the leaves, the tooth offers a considerable resistance to the recoil of the pinion, and rapid wear of the acting surfaces is the necessary consequence.

1126.—The designs of depths for low numbered pinions proposed by Camus made the lead commence considerably in

advance of the line of centres for the reasons given above; but as recoil escapements have been replaced by dead-beat escapements, there is now no occasion to facilitate a recoil that does not occur, at any rate to the same degree, while it has become important to diminish the engaging friction.

1127.—To set any stationary body in motion, we know that a much greater force must be applied than is required to maintain this movement (**29**). This being remembered, and observing that with escapements of considerable recoil the train, even up to the centre wheel, is always sensibly in motion, whereas with a dead-beat escapement this train is alternately in a condition of rest and movement, it will be easily perceived that the motive force has a much greater resistance to overcome after each locking in the second case; and if this resistance is further complicated by marked adhesion between the surfaces and by engaging friction, a great part of the motive force will be dissipated.

1128.—To diminish the duration and the amount of engaging friction with low numbered pinions, the point at which the lead commences must be brought nearer to the line of centres; this is effected by reducing the thickness of the leaves and increasing the height of ogives. In that case the extremity of the point of the tooth will be somewhat above the position indicated by the epicycloid and the end of the lead will no longer be absolutely uniform, but this slight inconvenience is balanced by the advantage (so important with the dead-beat escapement) of the lead commencing as near to the line of centres as is possible without serious inconvenience (**1117**).

1129.—The preceding considerations are based on the supposition that the spaces and teeth of the wheel are of equal width. It will be easily understood that as the width of a tooth is increased its ogive will become more elevated because the epicycloidal sides will become longer; but since the width of a tooth may only exceed that of a space by a very small amount (for otherwise the teeth would be too thick and the centres would have to be set farther apart), most makers have, at any rate for the last twenty years, adopted teeth and spaces of equal width as the best average dimension.

We shall see in the following articles in what cases we can advantageously introduce modifications in this general rule.

PROPORTIONS FOR PINIONS WITH ENGAGING FRICTION: 6, 7, 8 and 9 leaves.

FOR DEAD-BEAT ESCAPEMENTS.

1130.—REMARK. In low numbered pinions up to 9 inclusive it is the general practice to make the width of the space twice that of the leaf or, in other words, two-thirds space and one-third plain. This proportion has been adopted for the last twenty years or more by skilful practical men as the lowest permissible thickness that will ensure due solidity for the leaves.

Moinet's 7 and 9-leaved pinions do not conform to these practical rules; but we shall assume that they do so agree, for if his minute difference is perceptible in a large scale drawing it is not so in the small pinion itself, and the division into thirds is very easily determined upon by an eyeglass so that rapidity and accuracy of workmanship are greatly facilitated.

6-leaved pinion.

1131.—A 6-leaved pinion should have one-third plain and two-thirds space, and the depth of a space should be rather more than half the total radius of the pinion.

The teeth and spaces of the wheel should be of equal width, and the height of the point should be slightly less than half the primitive radius of the pinion.

If these proportions are adhered to in practice, the lead will commence at a distance from the line of centres equal to half the thickness of a leaf, in other words, when the leaf is first touched by a tooth, this leaf will be equally divided by the line of centres, so that its middle point will be directed towards the centre of the wheel as indicated in fig. 2, plate XIV. (**1135**).

7-leaved pinion.

1132.—A 7-leaved pinion should have one-third plain and two-thirds space. The depth of a space is to be between three-quarters and one half of the total radius, that is to say about three-fifths of that radius.

The teeth and spaces of the wheel should be of equal width, and the height of the point a trifle in excess of two-fifths the primitive radius of the pinion.

With such proportions the lead will commence when about two-thirds of the width of the leaf has crossed the line of centres (fig. 4, plate XIV.). We thus see that there will be less engaging friction than with a 6-leaved pinion (**1135**).

8-leaved pinion.

1133.—An 8-leaved pinion should have one-third plain and

two-thirds space, and the depth of a space should be rather more than half the total radius of the pinion.

The teeth and spaces of the wheel should be of equal width and the height of the point between two-sixths and two-fifths of the primitive radius of the pinion.

The lead will not commence until about three-quarters of the thickness of the leaf has passed the line of centres (fig. 6, plate XIV.), so that the engaging friction is still less than with a 7-leaved pinion (**1135**).

9-leaved pinion.

1134.—A 9-leaved pinion should have one-third plain and two-thirds space. Depth of the space, half the total radius of the pinion.

The teeth and spaces of the wheel should be of equal width. Height of the point just over one-third of the total radius of the pinion.

The lead commences very near to the line of centres. We have not given a drawing of the 9-leaved pinion as it is very seldom employed; the omission, however, can be easily supplied by making a large scale drawing in the manner explained in article **1100.**

FOR RECOIL ESCAPEMENTS.

1135.—The proportions given above should be generally adhered to, for they limit the amount of engaging friction to what cannot be suppressed without giving rise to inconveniences that are greater than those we wish to avoid (**1128**); but, as has been remarked in article **1125**, there are cases where it is preferable to have rather less lead after the line of centres. In such cases, as with the verge escapement for example, the above proportions are retained for the pinion, and the wheel is modified in the manner explained below.

The teeth being somewhat wider than the spaces, the sides of the ogives are made of a suitable epicycloidal form. It results from this slight change that the ogive is not quite so prominent and forces the leaf to a less distance and the following tooth is touched sooner, that is to say the lead commences in advance of the line of centres by a fraction of the thickness of a leaf that is almost exactly double that indicated under the head of each pinion.

PROPORTIONS FOR PINIONS WITH DISENGAGING FRICTION.

10-leaved pinion.

1136.—The 10-leaved pinion (fig. 1, plate XIV.) is the first

of those with which we can always ensure that the lead commences on the line of centres. But this condition requires that the leaf be very thin, in other words that rather less than 1/3rd be plain and more than 2/3rds space; a proportion which Moinet gives in degrees, 11° for the thickness of a leaf and 25° for the width of a space.

The depth of a space of the pinion is about half the total radius.

The ogive of the tooth should be epicycloidal, and in order to ensure that its height is sufficient the teeth are made somewhat wider than the spaces.

The height of the ogive is rather less than 2/5ths the primitive radius of the pinion.

With such proportions the lead is uniform and takes place entirely beyond the line of centres, but there will be very slight freedom in the depth and the least inequality of the teeth may give rise to catchings. With a view to avoid this fault it is a common practice to give a somewhat greater freedom, and the lead then commences a little in advance of the line of centres, if not with all the teeth at least with those that are unequal; and we know that such teeth cannot be considered rare in horological mechanism. When solidity is necessary, as in the centre wheel of a watch for example, it will be found beneficial to replace a 10-leaved pinion, the leaves of which are always of necessity very thin, by one of 12 leaves, as with it the leaves are more solid and allow of sufficient freedom to ensure the engagement taking place, while the lead is, as before, after the line of centres.

1137.—When there is no objection to a small portion of the lead being before the line of centres while it is desirable to avoid too great precision at the engagement, at the same time maintaining the strength of the leaf, the pinion may be made 1/3rd plain and 2/3rds space, the width of the wheel teeth being reduced; but it is essential to have, as a minimum, the teeth equal to the spaces (when the height of the ogive will be somewhat reduced).

We would here observe that as the leaves of pinions increase in number, the lead becoming shorter and shorter (**1123**), the inconveniences met with at the termination of the lead, referred to as existing with recoil escapements (**1125**), become

gradually less sensible and thus the most important aim should be to avoid the existence of engaging friction.

1138.—*Observations relating to the* 10-*leaved pinion.*—The great accuracy required in a depth comprising this pinion, in order to avoid lead before the line of centres and to prevent drop, explains the fact that very few are met with that satisfy the theoretical conditions.

For some time past great numbers of watches have been produced in factories of only moderate note in which 10-leaved pinions are used freely, and their makers do not seem to even suspect that the unintelligent construction of these pinions altogether neutralizes the advantages that are anticipated from their being high numbered. With badly made 10-leaved pinions the engaging friction is equal in intensity if not in duration to that of an 8-leaved pinion, and they do not, like this latter, secure a certainty in the engagement.

1139.—The 10-leaved pinion is much employed for the centre wheels of watches. In that position the lead on each leaf continues for six minutes; if one portion of it is accompanied by engaging friction it is evident that there will be serious variations in the motive force transmitted to the regulator during this period.

1140.—Moinet observes that there is danger of catchings with a 10-leaved pinion if the leaves are made too thin. We think this must be a typographical error, for the only danger of a thin leaf consists in its fragility and the risk of distortion when it is hardened. If he meant that it is a mistake to make the leaf thin without a proportionate increase in the width of the teeth, he should rather have said that catchings are liable to occur when the wheel teeth are too thin; this is true, especially with very thin leaves for with such the rounding will be continued for a very short distance on either side, and the primitive radius is longer than it should be; the pinion in fact is too *large* although its total diameter is correct (**1122**). This case is very frequently met with.

11-leaved pinion.

1141.—The advantages that characterize the 10-leaved pinion are still more marked in the case of one with 11 leaves. With the latter it is impossible to make the teeth and spaces of the wheel equal without danger of the lead commencing before the line of centres, and the pinion can have rather more than

1/3rd plain and less than 2/3rds space; the ogives of the teeth may be slightly less than 1/3rd the primitive radius of the pinion.

As the 11-leaved pinion is very seldom used we have thought it unnecessary to illustrate this case, but a drawing can be easily made in the manner explained in article **1100**.

12-leaved pinion.

1142.—The 12-leaved pinion (fig. 3, plate XIV.) is in reality the first of those with high numbers that allows of the teeth and spaces of the wheel being equal without involving any doubt lest the lead should commence before the line of centres, and it at the same time gives a leaf of sufficient strength with the requisite freedom in the depth.

It should have two-fifths plain and three-fifths space, or the plain is to the space as two is to three; thus if the arcs covered by a leaf and space be taken together and divided into five equal parts, three of these will form a space and two will give the thickness of a leaf (fig. 3).

The depth of a space should be rather less than half the primitive radius of the pinion.

The height of the ogive will be two-sevenths the primitive radius of the pinion.

If all the rules are carefully followed out the fine point of the ogive will not take any active part in the lead and may therefore be gently smoothed off, but it is well to retain it as a precaution, and from the fact that the untouched tooth presents a better appearance.

Pinions of 14 leaves and higher.

1143.—Above 12, that is to say commencing with the 14-leaved pinion (fig. 2, plate XV.), since we neglect that of 13 for a reason subsequently given, the lead always commences on the line of centres; the teeth and spaces of the wheel should be of equal width, and the spaces of the pinion rather wider than the leaves.

With the exception of the 14-leaved pinion we have omitted to give drawings of any above 12 leaves, but this omission can easily be remedied by drawings made with neatness and precision, and the explanations already given should amply suffice to enable the reader to prepare such figures.

We would only add in order to facilitate this work, although some of the following remarks have already been made:

always commence from the point of first contact to divide the circumference of the wheel, and complete the teeth; then mark off, provisionally, the thickness of the leaves, a thickness that must finally be determined by the amount of freedom necessary between the tooth and the rounding of the leaf (*o n*); for it will be evident that on diminishing the thickness of a leaf along *n v x* the freedom gradually increases.

The non-acting portion of the ogive may be removed.

When the teeth are required to withstand a considerable pressure the bottom of the spaces is rounded as indicated by dotted lines B and C (fig. 2, plate XV.).

1144.—It has been seen from article **1124** that with high numbered pinions the engagement becomes more and more fine and that the necessity of such very great accuracy may occasion catchings, especially in case the pivot-holes increase. High numbers, moreover, have the objection that the teeth are too fine to be sufficiently rigid.

Watchmakers will do well to adopt the intermediate numbers, in other words the series of pinions of from 10 to 18 leaves.

GENERAL OBSERVATIONS.

On the advantage of ascertaining the point at which lead commences.

1145.—The point at which the lead commences is, as we have already seen, fixed for each pinion; but it must be remembered that its position relative to the line of centres is only invariable on the supposition that the pinion conforms to the theoretical rules, for the reader has already understood that a change in the proportions occasions a variation also in the position of the point at which the lead commences, so that this point is moved: (1) away from the line of centres when the pinion is too large by an amount dependent on this excess, etc.; and (2) towards and even up to the line of centres with low numbered pinions if they are too small, etc.

In the first case there will be increased engaging friction with catchings, etc.; in the second case the lead will be irregular and characterized by slipping, drops, etc.

On pinions of odd numbers.

1146.—No inconvenience need be anticipated from the employment of odd numbered pinions providing they are made with care in accordance with the preceding rules, notwithstanding that certain watchmakers hold a contrary and unfounded

opinion. The only objections that can be urged against them are: firstly, the numbers 7, 11, 13, being *prime* numbers, with no divisors but themselves and unity, present certain difficulties when an attempt is made to combine them with other numbers that must enter into the calliper of a watch; secondly, there is some difficulty in accurately measuring the total diameter of a pinion and of ascertaining by the eye whether the faces are directed towards the centre, an inconvenience that is not experienced with an even number of leaves where there are always two leaves opposite to each other, and any error in the direction of the faces is immediately sensible to the eye.

1147.—The extreme diameter of an odd numbered pinion is measured on a thin plate perforated with holes, by introducing a small tapered scale rounded at its edges into the hole that the pinion enters without play. Then mark the point to which it descends, and the width of the rule at this mark being measured with care gives an exact determination of the required diameter.

On the importance of varying the form of the epicycloid to correspond with the diameter of the wheel.

1148.—The epicycloidal face of the ogive that engages with a given pinion varies slightly when the number of teeth of the wheel is changed, and this of course varies in diameter if the *pitch* is maintained the same; but this difference can be considered negligeable so long as the number is not varied from that originally fixed upon by more than a third or a quarter. Thus fig. 3 (plate XIV.) would give the proportion for the teeth of all wheels of from 50 to 120 teeth engaging with a pinion of 12 leaves.

If the pinion remain the same, the ogive will be a little more straight and elongated with a wheel of 120 teeth and the converse will be the case with a 50-tooth wheel, comparing in each case with figure 3.

1149.—If it be desired to ascertain the position of the pitch circle of a completed wheel, it must be remembered that the first part of an epicycloid is hardly at all curved and is almost a continuation of the straight face of the tooth, so that if the primitive circle be assumed to pass through the points at which the curvature of the ogive becomes somewhat marked, it will be too large.

Drawing a pitch circle on the teeth will materially help in ascertaining the correct pitching of the depth.

CHAPTER V.

SUMMARY OF THE CAUSES OF STOPPAGE AND VARIATION OCCASIONED BY BAD DEPTHS.

Practical Examination.

1150.—The details into which we have already entered will abundantly suffice to enable every intelligent and careful reader to understand the conditions that should characterize each transmission of movement by means of toothed gearing, and to discover the causes that may render this mode of transmission irregular and occasion wear of the acting surfaces, as well as those which may so far interfere with the action as to stop the movement of the machine.

This chapter then can only be regarded as a supplement to those that precede it, and we must here confine our attention to a brief enumeration of these causes and of the characteristic signs that indicate the existence of each particular fault and to directing attention to the exact points where correction is essential.

Depths are defective

1151.—(1) When the primitive diameters are not correctly proportioned, a fact which is expressed by saying that the pinion is *too large* or *too small;*

(2) When the centres of the mobiles are too far apart or close together, in other words when the pitching of the wheel and pinion is too shallow or too deep;

(3) When the teeth and leaves are incorrectly formed;

(4) When the leaves are too thick;

(5) When the teeth are too broad or too narrow;

(6) When the circumferences are unevenly divided or, in other words, when the teeth or the leaves are uneven.

Effects due to bad depths.

1152.—The detrimental effects are here classified in the same order as the causes that have produced them; so that each number corresponds to the same number in the previous article.

(1) If the pinion is too large, there will be a loss of force

caused by wear and catchings which will in time occasion the stoppage of the machine.

When the pinion is too small the greater part of the lead will be characterized by slipping and drops; both causes of wear and loss of force.

(2) When the depth is too deep, the back of the tooth rubs against the leaf or very nearly does so at the moment of its entering a space of the pinion, and there will be slipping and even a drop.

When the depth is too shallow the lead commences farther from the line of centres. There is a considerable amount of engaging friction with a tendency to a butting action, and the lead becomes an irregular and rapid slipping action.

(3) When the ogive of the teeth is too pointed, it is necessary to pitch the depth very shallow, and with short ogives the pitching must be deep. The inconveniences of such proceedings have just been indicated. Moreover an ogive that is too short (too much rounded on its faces) does not impel the leaf far enough if the depth is correctly pitched, besides which there will be engaging friction; if the pitching is too deep the engaging friction will become less but the lead will be characterized by irregularity in the rate of movement and drops.

The measurements adopted for determining the sizes of pinions (**1108**) have been calculated for the case in which the point of the leaf is rounded in a semicircle; it follows therefore that, if these measurements are applied to pinions with leaves of barleycorn form, the pinion will be found too small as regards its primitive diameter, and the lead, especially towards its conclusion, will be simply a prolonged slipping action or a drop.

(4) When the leaves are too thick the lead commences as much in advance of the line of centres as the thickness is in excess.

If with leaves that are too thick the lead commences at the proper point, the pinion is too small; the conclusion of the lead will only be a slipping followed by a drop.

(5) When the teeth are too wide, the freedom of the depth is insufficient. In order to avoid catchings it will be essential that the pitching be shallow.

If the teeth are too thin, the ogive cannot have the requisite height or form; the lead becomes nothing more than a suc-

cession of engaging frictions, irregular slippings and drops. It is usually more easy to secure a passable depth with the teeth of the wheel too thick than when they are too thin.

(6) Inequalities either of the teeth or leaves may arise from distortion in the hardening or from want of truth in the wheel-cutting engine, rounding-up tool, etc., or from the wheel or pinion not being true on the pivots, and these inequalities may give rise to any of the faults above indicated. At some places the depth will be shallow, at others deep; some teeth will give rise to considerable engaging friction while others cause a drop, etc. These faults, which are due to bad workmanship, may be detected by the fact of the lead commencing nearer to the line of centres with some teeth than others.

1153.—Whichever of the above cases is found to occur, there will always be an irregular transmission of force and therefore a loss of energy, because the useful effect is less than it might be. The motor experiences an excessive resistance and, in order to overcome it, must have greater energy, the effect of which will be a more rapid wear of the acting surfaces.

A train that is marked by several of these faults cannot fail to transmit irregular impulses to the balance, and the detrimental effect of such on the timing is well known. If the train be allowed to run down it will do so noisily and with constraint.

To practically examine a depth.

Summary of the characteristic signs of faults.

1154.—Five principal points have to be considered when examining a depth, namely; the *engagement*, the *commencement of the lead*, the *freedom*, the *lead* generally and especially its *termination*.

(1) The *engagement* should take place without straining and there should be enough freedom between the back of a tooth as it enters a space and the next leaf of the pinion.

(2) The *commencement of the lead* should take place on the line of centres or, in the case of low numbered pinions, at the distances from this line indicated in chapter IV. for each kind of pinion.

If the lead commences too far from this line, it proves that

one or more of the following faults exists; pinion too large, leaves too thick, ogives too short or too much rounded, pitching too shallow, etc.

Should it commence nearer to the line of centres than the required amount one or more faults may exist, such as: too small pinion, ogive too much elongated, pitching deep, etc.

(3) In whatever position the tooth be, it must always be able to move with a certain amount of *freedom* between the leaves.

(4) Throughout the *lead* the acting surfaces should always develope uniformly from each other. By holding the depth up to the light and examining by means of a powerful eyeglass, it is easy to ascertain whether one surface slides too rapidly at any period, as would be the case with a badly formed ogive or with a pinion of the wrong size or when the pitching is too deep or shallow. For it will be well to remember that opposite causes may often give rise to identical results. Thus if a depth be pitched much too deep or too shallow it will in both cases cause a stoppage through butting action.

(5) It is very important to observe with the most minute care the *termination* of the lead. If there be any slipping or drop or if the tooth leads by its point, we may be certain that one or several faults exist: the pinion is of wrong size, pitching too deep, ogive too short or badly formed, etc.

1155.—Examiners and manufacturers that are thoroughly conversant with the principles that have been developed above, as well as with the practical observations that supplement them, can arrange for themselves a rapid and at the same time exact method of examining which, after a short time, will naturally lead to rapidity and certainty in their work, such as cannot be arrived at by adopting the so-called practical methods of examination, for these are not founded on any precise principles and therefore compel those that adopt them to work in the dark.

Verification by touch.

1156.—Watch repairers that examine watches and receive but insufficient payment for their work, may proceed with

greater rapidity in the examination by accustoming themselves to *feel* the depths.

A point of pegwood is held in one hand and pressed on the top pinion pivot while the other hand turns the wheel gently by means of a second pegwood point. Several consecutive turns of the wheel in the direction in which it is intended to go will indicate whether the lead takes place smoothly and without scraping, catchings, etc. Carefully feel the amount of freedom in each successive position of a tooth.

Now make a second verification of the depth on the depthing tool; the two examinations will thus control each other. It will be well to perform the double operation for some time, in other words until the hand has attained to great sensitiveness and the workman is perfectly cognizant of the amount of freedom necessary to ensure the proper action of the parts; when this point is arrived at he can do without the depthing tool and have no hesitation in testing the depth by mere touch.

We recommend the practice of verifying on the tool for some time concurrently with the other method because without a certain amount of experience one is easily deceived by shallow depths that are smooth to the touch and which, while not actually causing the watch to stop, interfere seriously with the timing.

1157.—For testing clock depths the pegwood is nearly always useless because the fingers can gently press against the axes and allow or suppress the freedom of the pivots in their holes at will while at the same time giving motion to the wheels.

If necessary the workman can make openings or *lanterns* to enable him to examine the depth with an eyeglass. These afford an excellent mode of verifying but should only be employed when the lanterns can be made without impairing the solidity of the piece perforated or the good arrangement of the whole, or increasing the difficulty of repairs that may have to be made in time.

At the pivots of the extreme mobiles of a watch, which only require a very small quantity of oil, it would be well to make the oil cups no larger than is necessary and to leave the external face of the jewel flat. It will then be possible to follow the action of the teeth and leaves by looking through the jewel with an eyeglass, the plate resting flat on the bench or being held vertically in the air to a cross-light.

To remove the causes of Stoppage.

Reducing the size of a pinion.

1158.—In high class clock and watch-work a pinion that is not of the exact size to secure a good depth must be replaced.

In the more ordinary horological mechanism the watchmaker, especially in the country, will usually be under the necessity of retaining a pinion although he may be satisfied that it is defective either as regards form or dimensions.

As a rule in such cases:

If the pinion is slightly *large* the pitching must be *deep*, in other words the point is found at which the tooth impels the leaf to the greatest possible distance without danger; for the engaging friction of the lead has to be diminished as much as possible.

Or the size of the pinion can be reduced in the manner indicated in article **1122**.

When a pinion is somewhat *small* there is not much danger of butting, but the drop will be considerable; this fault can be reduced by pitching the depth rather *shallow*.

Or, if preferred, the primitive diameter of the pinion can be increased by the method referred to in article **1122**.

1159.—A slight inequality in the teeth of the wheel and another in the leaves of the pinion, resulting from bad polishing, or a wheel or pinion out of truth, which is very common, occasionally and at long intervals combine together and give rise to stoppage that causes great trouble to watch repairers. This is especially the case with a 10-leaved pinion, and, in general, with all pinions that are at all too large. By slightly diminishing the pinion the cause of stoppage will be removed.

This can be accomplished, in cases where the pinion cannot be replaced or the wheel removed, by the following method. The total diameter of the pinion is first reduced on the turns, as indicated at *a b* (fig. 13, plate XV.) and the sharp corners of the leaves are rounded off as well as possible by scraping in the direction of their length with the cutting edge of a fine, well sharpened graver. The pinion should rest in a hollow made in a wooden block, so that the pressure of the graver may neither strain the wheel nor loosen the riveting. Or the manipulation might be rendered still more easy by waxing the wheel to a brass disc or a ferrule.

Having done this, the wheel is set between the two centres of an instrument arranged for the purpose (a depthing tool for

example) and between the other two centres is placed the axis of a wooden polisher *cf*. The latter pair of centres is unclamped, but they are united by a rigid metallic arc that maintains them firmly, while it is possible to give a sliding motion to the polisher in the direction *fg*.

The remainder of the operation will be easily understood: a bow is placed on the ferrule of the polisher, which is then caused to rotate while engaging with the pinion, the wood being charged on its circumference with oilstone dust and oil; and the rotation is maintained while a sliding movement is given to the polisher as indicated by the arrows. The operation should then be completed by using another polisher charged with rouge.

The form and general arrangement may be varied if desired, but every watchmaker will be perfectly competent to adapt it to his particular requirements without further explanation.

1160.—Teeth that are rough to the eye or whose faces are not perpendicular to the plane of the wheel are often much improved by smoothing with soft charcoal. A small piece of it is used moistened with oil or it may be ground in oil and used on a brush. Only angles that are too sharp must be smoothed and rounded, and care should be taken not to distort the teeth. The method involves caution and intelligence.

CHAPTER VI.

VARIOUS KINDS OF DEPTHS.

DEPTH IN WHICH THE PINION DRIVES THE WHEEL.

Each mobile drives and is driven.

1161.—We have hitherto only considered the most general case, namely that in which the wheel leads the pinion; we now proceed to examine the special case of the pinion driving the wheel, or of each mobile being alternately driven and driving.

When the pinion leads the wheel the principle already enunciated still holds, that is to say *the primitive diameters are invariably proportioned to the number of teeth;* the following are the only changes made:

(1) The ogive of the wheel tooth is a semicircle (fig. 10, plate XIII.);

(2) The slight increase recommended (**1099**) for the primitive diameter of the leading mobile will in this case be on the pinion;

(3) The ogive, now formed on the leaves of the pinion, is indicated by the epicycloid traced out by the generating circle G D F (fig. 10, plate XIII.), whose diameter is equal to the primitive radius of the wheel;

(4) The leaves and spaces of the pinion are of equal width or very approximately so; the spaces of the wheel are rather wider than the teeth, according to the general rule, to which there are but few exceptions, that the driving mobile should have the spaces and teeth or leaves equal, and the mobile that is driven should have the former wider than the latter to an extent that depends on the amount of freedom necessary, etc.

1162.—It will be observed that, when this class of depth is made in accordance with the theoretical principles, it is characterized by an absence of engaging friction; but, in the case of a 6 or 7-leaved pinion, the engagement of the point of the ogive will be marked by a very minute interval of safety. If the freedom be sufficient to avoid this fault we are met by the difficulty of a slipping of the point of the ogive towards the end of the lead, and there will be a slight engaging friction. With an 8-leaved pinion and upwards these difficulties need not be anticipated; the ogive becomes gradually longer, and it may even be found necessary to remove that portion of it that takes no part in the lead and whose only effect is to interfere with the engagement.

1163.—We see that the practical rule to make the pinion somewhat large is utterly at fault, for the primitive diameter remains the same; it is only the rounding that is prolonged and changed in form. It is by no means rare to meet with pinions that satisfy this practical but erroneous rule, especially in the motion work; thus these pinions, with a primitive diameter that is too great and rounded when they should be formed with an ogive, are characterized by a lead that is all or nearly all accompanied by engaging friction.

When the depth is required to act in both directions, each mobile alternately driving and being driven, the pinion is formed as above explained and the points of the wheel teeth are epicycloidal ogives; but it is usually better in such cases to employ involute teeth.

TEETH OF A CURVED RACK.

1164.—Since a curved rack is nothing more than an arc

of a circle, or rather a portion of a wheel, such a depth will be included under the general conditions.

The following method is adopted in most cases:

First determine the distance D *s* (fig. 14, plate XV.) between the two centres, then draw the two radii D *a*, D *b* from the centre of the rack, giving the extent of its displacement, that is to say the space over which it is required to move, which will be represented by the arc *a c b*.

Half of this space, or the angle *c* D *b*, gives the length of rack that is indented. Assume that the pinion performs two complete rotations while the rack travels from *c* to *a*. The pinion then should have a number of leaves equal to half the number of teeth in the rack *c b*; let 6 and 12 be these numbers. Measure the angle *c* D *b* accurately in degrees, and it should always be an even fraction of 360°. As an illustration, let this angle be 40° and we shall find: if 40° contain 12 teeth and 12 spaces, the total circumference will contain 216 (24 × 9), for 40° × 9 = 360°, or an entire circumference. Thus if a complete wheel were required it would have 108 teeth.

The remainder of the operation resolves itself into determining the proportions of an ordinary depth between a 6-leaved pinion and a 108-toothed wheel, the distance between the centres being known (**1049**).

If the pinion were only required to perform half a rotation for the entire length of the rack, the number of leaves would be twice the number of teeth in the length of the rack; when the pinion only makes one turn, the two numbers will be equal; if it makes two turns the number of leaves will be half the number of teeth, etc.

TEETH OF A STRAIGHT RACK.

1165.—The teeth of a straight rack (fig. 10, plate XV.) differ from those just considered because in this case the movement is transmitted from a circle to a straight line or conversely. The curves employed will no longer be the same.

When the rack drives the pinion, the ogive of the teeth will be given by *the cycloid traced out by a generating circle whose diameter is the primitive radius of the pinion.*

When it is the pinion that drives the rack, the ogives of the leaves acting on the faces of the teeth must be the involute of the generating circle.

In either case the primary proportion will remain as before;

that is to say, *the length of rack is to the geometrical circumference of the pinion as the number of teeth is to the number of leaves.*

If we compare the cycloid and epicycloid that are traced out by the same point of a generating circle when it rolls on a straight line and circle, we notice that the cycloid ascends less rapidly so that it will impel the leaf to a less distance, and thus there will be a greater proportion of the lead before the line of centres (fig. 10).

Contrate depths for watches.

1166.—Contrate wheels, as they are made in watches, do not give a uniform lead. They belong to the class of skew or bevel depths, that is to say those whose axes are inclined at an angle instead of being parallel (**1178**); hence it follows that not only must the pinion be conical, but also the ogives of the teeth, having surfaces of double curvature, should be parts of spherical epicycloids: conditions that are quite impracticable on a small scale.

It is therefore considered that a sufficient approximation to the uniformity of the theoretical lead is secured by making the teeth resemble those of a straight rack, to which they are somewhat analogous, and to introduce certain modifications for the purpose of facilitating the movement, such as teeth that are much thinner at the top than at the bottom (measuring the thickness at the outside), a pinion that is slightly small, the pitching rather shallow, spaces wider than the teeth, and if possible the latter inclined outwards and rounded crosswise on the acting faces; the pinion leaves should by preference be barley-corn shaped; the commencement of the lead with a 6-leaved pinion is at least 2/3rds of the thickness of the leaf in advance of the line of centres, etc.

These modifications are evidently made with a view to facilitate the recoil of the escape-wheel from the verge, by diminishing the amount of lead after the line of centres. It will be obvious that the more it is continued beyond this line, the more the tooth butts against the leaf on reversing the motion, occasioning a resistance to the recoil. The escapement is thus overpowered; it regulates with difficulty, and rapidly becomes sluggish in its movement.

The facility of recoil is one of the signs of a good contrate depth employed in a train that is governed by a recoil escapement.

By increasing the number of leaves of the pinion the recoil becomes less and less constrained, since the duration of the lead is reduced.

The article on skew depths (**1178**) will give the reader some further information; it will also enable him to form and pitch his contrate depths to the best possible advantage.

INTERNAL DEPTHS.

1167.—When it is required that two parallel axes turn in the same direction, an internal depth may be resorted to.

Fig. 7 (plate XV.) represents such an one: the pinion, having straight faces, is driven by a wheel whose points are traced out by the rolling of a small generating circle *c* inside the pitch circle of the wheel (**1038**).

This class of depth is very rarely used by watchmakers, and we would refer the reader to special treatises for fuller details on the subject; we will only observe that they give rise to friction of somewhat less extent when the wheel drives the pinion, as indicated in fig. 7, than when this same pinion is driven by a wheel having the same pitch circle but external teeth.

TO DRAW INVOLUTE TEETH.

1168.—Let circles described with the radii R *o*, P *o*, so that they touch on the line of centres at *o*, be the pitch circles of the wheel and pinion (fig. 12, plate XV.).

Through *o* draw a straight line *f o t v*, which will be a secant to the two circles. From the centres P and R draw perpendiculars P *m*, R *t* on this line; and with P *m* and R *t* as radii describe two circles concentric with the first circles. These circles will bear the same proportion to each other as the pitch circles do.

As the contact of the two teeth has to occur on the portion *m t* of the line *f o t v*, draw from a point taken on this portion of the line, *s* for example, the involute *c s d* of the circle P *m*. Through this same point *s*, which is the point of contact, trace out the involute *g s h* of the circle R *t*.

The curves *c s d*, *g s h*, determine the sides of the points of the leaves and teeth respectively.

If it be required that the drawing represent the depth at the moment of contact on the line of centres, it is only necessary to draw through *o* two curves parallel to those previously drawn; and then commence from *o* when subdividing the wheel and pinion.

The effective height of the points should be indicated on the drawing in order that the spaces may not be deeper than is necessary.

In depths of this class the lead takes place partly before and partly after the line of centres. The two mobiles can alternately drive and be driven without requiring any change.

1169.—*Observations.*—Engineers, when they trace out teeth of this form, incline the line *f o t v* at an angle of about 75° to the line of centres. Some prefer to adopt the following method: measure from the point *o* on the circumference, *z*, an arc equal to twice the pitch of the wheel;—assume this to be *o z f* (exaggerated in order to render the details more intelligible). Through the point *f* thus determined and *o* draw a straight line, which will be the required secant.

After the pivot-holes have become somewhat enlarged, involute depths still work very well and with sufficient uniformity in the lead and pressure (**1083**).

LANTERN PINION DEPTHS.

1170.—The two discs that maintain the cylindrical bars of a lantern pinion in position are termed *frames* or *plates*, and the bars, which take the place of leaves, are known as the *rounds*.

A lantern pinion depth (fig. 5, plate XV.) must be made in accordance with the invariable law that the primitive diameters are proportional to the number of teeth. The pitch circle of the pinion, passing through the centres of the rounds, *v*, *r*, *a*, *b*, *t*, etc. is also the generating circle for the points of the teeth.

The plan of a depth of this description is based on the following theorem:

Let P (fig. 6, plate XV.) be the pitch circle of a lantern pinion whose rounds are as fine as the points that represent them, and let R be the pitch circle of the wheel. If the two circles are placed in contact at the point *a*, and P is caused to roll on R, this point *a* will trace out the epicycloid *a o b*. But since this curve has been traced out by the point itself it necessarily follows that the straight line *x y* is a normal both to the curve and to the fine pin *a*, and it further follows, from the previous considerations, that if this fine pin were fixed in the circle P and impelled by a piece of the form *a o b* projecting from the circle R, the disc R would drive the disc P with a uniform velocity and force.

Now suppose that with the point *o* as a centre we describe the circumference of the round *n c*, and from the point *c* draw the curve *c d* parallel to *a o b*; we could prove, for the reasons already explained, that the straight line *x y*, which is normal to the point *o* and to the epicycloid *a o b*, is also normal to the curve *c d* and to the cylindrical face of the round; hence it follows that, if the pin *a* have the diameter *n c*, and the ogive on the disc R be of the form *c d*, this ogive will drive the round in the same manner as the epicycloid *a o b* would drive a fine point or round *o*.

To draw a lantern pinion depth.

1171.—After having determined the primitive diameters that are proportional to the number of teeth, draw the two pitch circles; let these be D *a* D′ and *v r a b t* (fig. 5, plate XV.) tangential to each other at the point *a* on the line of centres. This point will be the centre of a round; and, after having marked it out, it will be easy to fix the centres of all the other rounds.

The *pitch* of the wheel is known (since it is the length of the circular arc *a b*); if not known it might be calculated, and then take half, which will, in the most general case, give the width of a tooth, since the teeth and spaces are equal; the other half of the pitch will represent the diameter of a round, *c d*, plus the amount of freedom necessary to ensure the proper engagement of each tooth with the rounds, that is the interval *d h*. The amount of this freedom will be diminished as the number of teeth in the mobiles is increased. It can be approximately determined for the several cases by measurements taken from the figures on plates XIV. and XV.

When it has been ascertained we know the diameter of the first round *c d*; this can be drawn in, and then all the others, *v*, *r*, *b*, *t*, etc., in succession.

Commencing from the point *a*, trace out the epicycloid *a f*, and, with a radius equal to that of the round, describe the series of small circles *i*, *i*′, *i*″, etc., having their centres on this epicycloid; then find by trial the centre from which a circular arc can be traced tangential to all these small circles. This arc, *c n* (fig. 5), is the side of the ogive, and it only remains to subdivide the pitch circle and fill in the teeth.

The final contact of a tooth and round occurs at *p*, for the

succeeding tooth then just comes into action; this fact gives a means of determining the useful height for the ogive.

1172.—*Observations.*—In writing on horology watchmakers that only concerned themselves with watches and timepieces have condemned lantern pinion depths without exception.

On the other hand, those who occupy themselves with the manufacture of turret clocks, and have had experience of lantern depths, continue to employ them with some advantage.

This contradiction can be very easily explained.

It is objected to the lantern depth that:

(1) When the wheel impels the pinion;—the two acting surfaces do not develope from each other, and a portion of the lead takes place before the line of centres.

(2) When the pinion drives the wheel;—the lead is characterized by engaging friction and a butting action which very rapidly destroy the surfaces that come in contact.

Theoretical men, and mechanicians who followed in their track, considering that in the great majority of machines the mobiles will be called upon alternately to drive and be driven, have concluded that this form of depth is *incomplete.*

This consideration loses nearly all its force in the case of turret clocks, where wheels that are high numbered drive pinions of at least eight or nine rounds and always in the same direction.

We may dismiss the very rare case of the pinion driving the wheel; in such a case the lantern would be useless and a solid pinion with leaves terminating in ogives would be essential; but let us compare two depths with the same number of teeth and primitive diameters, but one a lantern pinion depth and the other with a solid pinion, the leaves having straight faces; it will be seen that:

The lantern depth is objectionable in that:—there is rather more lead before the line of centres and a less extent of surface acting on the round, which may be assumed to be stationary.

It is advantageous because:—the pressure towards the end of the lead is less since it acts through a longer virtual lever, and the pressure is less variable throughout the lead.

We therefore conclude that if the surface of each round is accurately parallel to the face of the tooth and the width of this latter is proportional to the pressure applied at it (**40**), the total friction in the two classes of depths may be regarded as equal.

Experience has confirmed this conclusion, and its accuracy

can be easily proved by an examination of well-made clocks. Among those we have examined, several that were made by the celebrated family of Lepaute, some with fixed rounds and others in which they were movable, only showed very slight signs of wear. although evidently very old and much used.

In conclusion then, when we remember the ease with which a perfect lantern pinion can be made without any risk of distortion in hardening, as is the case with the solid pinion, we must come to an opposite conclusion from that usually current amongst watchmakers, namely that a lantern pinion, if well made and employed with judgment, possesses many recommendations for ordinary clocks of large size.

The employment of loose rounds, that is rounds that were formed into pivots at their extremities, has been very generally abandoned notwithstanding the manifest reduction in friction that they secured. When the pivot holes became enlarged through wear, the repair of the pinions was very difficult and costly.

1173.—It is not customary for manufacturers to harden the rounds because they have never been able to detect more than a very slight amount of wear, which was negligeable. Yet we cannot help thinking that if the rounds were hardened at their acting surfaces they would ensure still greater certainty in the action.

WORM-WHEEL DEPTH.

1174.—If the pitch of a screw be the same length as the pitch of a wheel, which is measured along the primitive circle, it is manifest that one turn of the screw will only advance the wheel by one tooth, whereas one rotation of the wheel will cause the screw to rotate as many times as there are teeth of the wheel. This simple mode of causing one axis to perform a much greater number of revolutions than another by substituting a screw for the ordinary pinion, has recommended itself to some watchmakers for use in horological trains. Their attempts have always failed because the movement of the screw involves a considerable pressure in a direction different to that in which the rotation occurs, adhesion effects are occasioned by this pressure that are very variable, all the more so when the screw is inaccurately made, when, as is by no means rare, it is distorted in the hardening, and when the velocity of movement is reduced. Besides this nearly the entire pressure of the wheel takes place along the axis of the screw and is supported by the

extremity of a pivot; it is thus concentrated on a small area, which soon shows signs of wear. If in order to diminish this effect the pivot be made large, the resistance opposed to the movement will be augmented very materially with the least thickening of the oil.

It is advisable that as far as possible the action of the wheel be opposed to the weight of the mobile that carries the screw; the friction of the pivot that is due to the weight of the mobile will thus at any rate be avoided. But this advantage is of minor importance compared with the condition that the pressure must not lift the mobile, since there would then be a displacement and loss of time.

1175.—*To draw the depth.*—Let P be the pitch of the screw (*a b*, fig. 9, plate XV.) and N the number of teeth in the wheel; the pitch circle of this latter will therefore measure P × N.

The primitive diameter of the wheel will be obtained from the proportion

$$3{\cdot}1416 : 1 :: P \times N : x.$$

Half of this value of x is the primitive radius of the wheel.

With the radius D *c* so obtained (fig. 9) describe the circle *j c s*. At the point *c*, erect a perpendicular *g f* which fixes the *pitch line* of the screw.

Divide the wheel into N equal parts commencing at the point *c*, the length of each arc *c s*, etc., being equal to the pitch of the screw *a b*. Draw in the teeth, equal in width to the spaces, or, if the thread is very fine, somewhat wider than the spaces; the sides of the ogives will be traced out by a cycloid generated by a circle of diameter D *c* rolling along the straight line *g f*. The thickness of the thread must be reduced in order to give the requisite freedom to the depth.

For each revolution of the screw with a given wheel, one of its teeth will pass if the screw is single-threaded; two if double-threaded, etc. The velocity of the wheel increases then as we increase the number of threads, and *vice versa*, but the wheel will lead the screw with a less effort when the threads are numerous; or, to speak more accurately, as their inclination becomes more and more rapid.

These details, which will be rendered more complete by a few further particulars, will suffice in the majority of cases. A reader that is anxious to study the subject thoroughly must have recourse to special treatises in which it is considered.

The acting faces of the teeth should be inclined to the flat of the wheel to the same extent as the thread is inclined to the axis of the screw. When this inclination is slight the screw will always turn the wheel with ease, but such will not be the case if the wheel is required to drive the screw. An amount of resistance will be opposed that will prevent its movement; in such cases, therefore, the inclination should be considerable, not less than 45°.

1176.—When a screw has several threads the distances between them are equal fractions of its pitch, or the height of one helical turn of a thread. This pitch will cover as many teeth and spaces of the wheel as there are threads of the screw (**1041**).

If the threads are numerous and very much inclined, the pitch becomes very great, and the axes, which were crossed with a single threaded screw, will become parallel and in the same plane. The wheel will then only act on very short portions of the thread by successive contacts, and it becomes a helical depth (**1082**).

1177.—*Practical details.*—The drawing only represents one point of the screw as being engaged with the wheel; but when it is necessary to overcome a considerable resistance, it is usual, in order to have greater surfaces of contact, to deepen the spaces of the wheel by the help of the screw itself. To avoid the risk of damaging the one that will be finally employed, another identical with it should be used; the front edge of the thread should be made cutting and then be caused to act against the circumference of the wheel, the two being gradually brought together as the teeth are formed, until the screw no longer acts.

This method can be adopted either for cutting the wheel in the first instance or only to deepen or incline its teeth; but if the wheel is intended to drive the screw, the thickness of the thread must be subsequently reduced, and it should be polished after having ensured the requisite freedom of the depth and re-adjusted the curves where necessary. If the surfaces of contact are too great, the resistance, as well as any slight inaccuracies of construction, would be more marked. On the other hand large surfaces are preferable when the screw drives and considerable force has to be overcome.

An endless screw engaging with the dividing plate of the wheel-cutting engine makes it a much more serviceable tool,

and by adopting the method here given any watchmaker can easily adapt one when required.

SKEW OR BEVEL DEPTHS.

1178.—Figs. 8 and 11 of plate XV. represent two skew depths, that is depths in which the axes of the two mobiles are not parallel.

The drawing, and more particularly the practical construction, of a depth of this description present difficulties of several kinds; they are very rarely employed in horology except for driving mobiles whose action is comparatively unimportant, and watchmakers are therefore usually content if they secure a lead that is fairly smooth without troubling to secure strict uniformity of force or velocity.

The angle between the two axes being known, draw two lines *a d*, *a b* (fig. 8, plate XV.), so that *d a b* is equal to this angle.

The numbers of teeth have been already fixed upon and suppose that the positions to be occupied by the wheels are also known.

Taking care not to go beyond this position, mark off from the point *a* along the axis of the smaller mobile a length *a s*, and on the axis of the larger wheel a less length *a c*, which must be to each other as the number of teeth in the large wheel is to the number of teeth in the small wheel.

At *s* and *c* erect perpendiculars cutting each other in *n*. Join *n* and *a*, and measure off *s g* equal to *s n*, *c f* equal to *c n*, and join *g* and *f* with *a*.

Having indicated the thickness that it is desired to give to the larger wheel by the parallel straight line *i j*, the thickness of the smaller wheel will be determined by the line *i r* drawn through the point *i* parallel to *n g*.

The triangle *n a f* represents the section of a rotating cone which drives by simple contact another cone whose section is the triangle *n a g*. The zones *n i j f*, *n g r i* are thus the primitive zones of the required depth. It remains for us to subdivide them and to determine the forms of the points of the teeth.

As the faces of the teeth are all directed towards *a*, their thickness diminishes in that direction as well as that of the spaces. The form of the point will be determined by the surface generated by the line of contact of a cone (whose diameter is equal to the radius of the mobile that is driven) rolling on the surface of the cone that drives (as the cone *m u v*, fig. 11,

plate XV.). These points, as will be evident, are very difficult to make, since their acting surface has a double curvature, that is to say it is generated by a spherical cycloid.

Hence it is usually considered sufficient in practice to cut the spaces of equal width throughout, and to make the curve *x z* (fig. 11) the same as in a straight rack that drives a pinion. The tooth is then cut out to this curve or to the circular arc that most nearly approximates to it, directing the curved face of the tooth towards the centre, that is to say so forming it that if a straight rule be applied at different heights of the point (as *s z* fig. 11), it will always be directed towards the centre at which the axes of the wheels would meet if prolonged.

DEPTHS WITH SEVERAL TEETH IN ACTION.

1179.—In the various depths that we have hitherto considered, as soon as a fresh tooth of one mobile engages with a tooth of the other, the tooth immediately preceding it, which was leading, ceases to engage. It is however possible by the adoption of certain forms of teeth to devise a depth in which several teeth of the wheel that leads are always in contact with a similar number of teeth of the wheel that is led.

This class of depth might perhaps be of some service in horological mechanism, more especially in the construction of keyless work. Watchmakers will find them discussed in the larger mechanical treatises, such as Hachette's *Traité élémentaire des machines* or Willis' *Principles of Mechanics.*

ARITHMETICAL CALCULATION OF A DEPTH AND SMALL SCALE DRAWING.

TO REPLACE A LOST MOBILE.

1180.—The arithmetical calculation of a depth is exceedingly simple and any difficulty will be removed by the various details and explanations that precede, especially in the case of depths such as those considered in chapter IV.

When it is required to ascertain the proportions of any other kind, a large scale drawing should be made showing different positions of the tooth throughout the lead, and, after having calculated by the well-known geometrical methods and tables the exact values in millimetres and decimals of the circumferences, diameters, radii, chords, etc., it is easy to deduce the required proportions and to reduce the dimensions to what is necessary. The practical applications of these measurements will present no difficulty if they are taken with some kind of

proportional compass and the various micrometers that are described in this work.

We will here only give a simple practical application, explaining the method to be adopted in order to ascertain the dimensions of one of the mobiles of a watch train when it is very much damaged or has been lost.

Two cases present themselves: in the first not only the wheel may be lost but also the pinion that engages with it; in the second the wheel only is supposed lost. We will consider the two in succession.

To Recover the proportions of a wheel and pinion.

1181.—At the outset determine the number of teeth of the wheel and pinion since these numbers are not known. They must be so selected that, when forming part of the train of the watch, the number of vibrations per hour is such as to secure a satisfactory rate. (To calculate the number of vibrations,* **1057**.)

Take, as an example, an ordinary watch that wants the third wheel and the pinion that engages with it and is concentric with the fourth wheel.

Assume that it has been ascertained by counting the vibrations of the balance that it makes just over 5 in a second; this will give rather more than 18,000 in an hour.

Let us further assume that on trying two numbers, say 60 for the wheel and 8 for the pinion, the vibrations as obtained by calculation are 18,200; it follows that 60 and 8 are the two numbers that should be adopted.

Measure accurately on the plate of the watch (by means of compasses and micrometers that go to fractions of millimetres) the distance between the centres, that is the sum of the primitive radii. We will take it to be 7 millimetres, or rather, for facility of calculation, 70 tenths, which is of course the same thing; and state the proportion as follows:

The sum of the numbers of teeth (60×8) is to the sum of the primitive radii (70), as the greater of the two numbers (60) is to the greater primitive radius, x (the unknown quantity, **1049**):

$$\text{or } 68 : 70 :: 60 : x \ldots x = \frac{60 \times 70}{68} = 61{\cdot}7$$

* It is possible to ascertain the number of vibrations per hour when we possess the balance of a watch or the pendulum of a clock by making it vibrate side by side with a comparison balance, such as those described in paragraph **430**, or with a known

So that x is 61 tenths and 7 hundredths of a millimetre. Since the mobile that leads should be slightly large (**1099**), the small fraction $3/100^{\text{ths}}$ of a millimetre may be added, and we obtain 62 tenths as the primitive radius of the wheel; and, subtracting this number 62 from the distance between the centres (70), there remain 8 tenths as the primitive radius of the pinion. Doubling these two radii gives geometrical diameters of 124 and 16 respectively.

If we take 3 as the height of the ogive of the tooth (**1133**), adding twice this amount to the primitive diameter gives a total diameter of 130 for the wheel (**1043**).

We have been performing the above calculations by means of tenths of millimetres, so that it is only necessary to separate the final figure in both numbers by a decimal point (which amounts to dividing them by ten) to obtain the values in millimetres. This, then, gives 12·4 millimetres as the primitive diameter of the wheel, and exactly 13 millimetres as its total diameter.

The total diameter of the pinion is easily ascertained, for we know that it is given by adding the thickness of a leaf at the commencement of the rounding to the primitive diameter, if the rounding is a semicircle. If micrometers and the gauges described subsequently are employed, this method gives a result of sufficient accuracy while it does not present any difficulty.

1182.—These measurements will enable us to re-make the depth, an operation which will be further facilitated by drawing a plan of the exact size.

To do this take a well smoothed plate of brass and mark the centres on it, taking them from the movement by the aid of the depthing tool. Open the sliding vernier compass to the primitive radius of the wheel, and from one of the centres, with this radius, make a small mark a (fig. 10, plate XVI.); then, setting it to the total radius, make another mark b; take a small compass, such as that shown in fig. 5, plate XVI., and carefully trace out the two circles a and b, following the compass with an eyeglass. The primitive circle of the pinion is tangential to a and may next be drawn; and, in order to obtain the total diameter of the pinion, measure the thickness of a leaf by the

pendulum. In the absence of such assistance the balance is set in motion and its vibrations counted as explained in **432** and the following articles. After a little practice it is easy to ascertain the number in a minute or half minute; and from this the number in an hour is at once calculated.

micrometer (chapter IV.), and half of this, added to the primitive radius, will give the total radius of the external circle *c*, that is, the size of the pinion. The plan of the depth is thus accurately obtained.

1183.—*Observations.*—If after the depth has been renewed the number of vibrations of the balance is not the same as formerly, so that there is either a loss or gain on the rate, this shows that the lost mobiles were not well proportioned; but as a rule it will only be necessary to slightly alter the length of the balance-spring to correct this error.

The small drawing compass (fig. 5, plate XVI.) should be employed only when the centres are pointed and not drilled through, for in the latter case its point would be liable to set obliquely in the hole.

To recover the proportions of a wheel only.

1184.—When only the wheel is wanting and we possess the pinion that engages with it, the total diameter of the pinion must be accurately measured, and the thickness of a leaf deducted from this in order to obtain the geometrical diameter. Subtracting half this amount from the distance between the centres, the remainder will be the primitive radius of the wheel. The proportion is then stated:

Primitive radius of the pinion is to *primitive radius* of the wheel as the number of leaves is to x, the unknown number of teeth of the wheel.

When the value of x is obtained we know the numbers of teeth and the primitive radii of the mobiles. From these values all the others can be obtained, as was done in the previous case (**1183**).

We will confine ourselves to the two applications here given in connection with this subject; for, whatever case presents itself, it can always be resolved into finding the two primitive diameters that are proportional to the numbers of teeth of the mobiles, or conversely.

Optical method suggested by Moinet for verifying the form of a tooth.

1185.—In order to judge whether the tooth approximates sufficiently in form to the large scale drawing, the latter is held at the level of the eye and in a vertical plane; now, closing one eye, place an eyeglass to the other and examine a tooth of the wheel, which is held at the focus of the glass and in front of the

drawing; then open the closed eye to see the drawing, so that the two eyes will at the same time see two images, which can be easily superposed. By moving the eye to or from the drawing the sizes of the images can be brought to agree so that the points of difference that have to be corrected can be observed.

Instruments for demonstrating the principles of depths.

1186.—For schools and public lectures, models of depths of very large dimensions are employed. These instruments show very clearly the manner in which depths act but they do not serve to explain their principles. It is better to employ for practical demonstration, arrangements similar to those which we exhibited in the Exhibition of 1867.

MOTORS.

General Considerations.

1187.—The motors employed to drive the mechanism of horology are of two kinds: weights and springs.

The first owe their power to gravity or heaviness, the others to the elastic reaction of bent metallic bands.

1188.—It is preferable that the action of a motor be uniform throughout its entire period of action; but, if it is impossible to satisfy this condition, the energy of the motor must increase or decrease with very great regularity. Nothing so much interferes with uniformity of timekeeping as sudden variations in the motive force; and it is important to always remember that an escapement cannot exercise its correcting influence outside certain limiting values of the motive force, which values are often very near together. Moreover, even between these limits, any sudden change in the force that impels the balance will occasion irregularities in the movement, etc., *in consequence of the resistance due to inertia*, which the balance-spring, whether isochronal or not, will be incapable of counteracting, because *it also is subject to the laws of inertia.*

1189.—Thus whatever care be taken in the selection and adjustment of a motor, the advantages anticipated from its use will be nullified if the force is transmitted through bad depths.

Increase in the friction, variations in the velocities, etc., which are added or subtracted as we pass from one depth to another, are quite sufficient to make the impulses communicated to the balance irregular, however uniform the action of the motor may be.

WEIGHT MOTOR.

1190.—A suspended weight that descends uniformly (if the velocity of its movement varied the power exerted would also alter) is a motive force that always remains the same, if we neglect the slight increase in the weight that is due to the uncoiling of the cord, an increase which may be considered inappreciable for a descent of one or two yards.

When, however, the weight is suspended by a chain or very thick cord, or when the fall is considerable, the increase in the force exerted due to the uncoiling of the cord or chain must enter into the calculation.

The fall of the weight must, as we have already seen, take place with perfect regularity in order that the force exerted may remain constant. Whenever its motion is characterized by jerks or shakes through the cord being elastic, when there are sudden contacts, variable adhesion, etc., these will all modify and decompose the power exerted by the motor.

The statical force exerted by a weight attached to one end of a lever capable of rotating on a horizontal axis will vary with the position occupied by the lever. It is a maximum when this is horizontal, and gradually diminishes to the vertical position, where the action of the weight consists solely in a pressure on the pivots of the axis (z, fig. 67, page 660).

If the lever move with different velocities, or if its angular movements are unequal, the dynamical force exerted will be very variable (**25**).

We would commend the study of these problems to the misguided seekers after constant force.

1191.—In large clocks and regulators the weight should be at a sufficient distance from the pendulum because, if they are close together, it may happen that, when the weight has descended to a distance equal to the length of the pendulum, the two contiguous weights influence each other by what is known as sympathetic action.

1192.—For a given weight the motive forces are to each other in the ratio of the radii of the drums on which the cords are wound.

With drums of different diameters it is possible to obtain the same motive force by employing weights that are inversely as the radii (ignoring the friction at the pivots, which is proportional to the weights and the diameters of these pivots, **133**).

The length of fall is necessarily proportional to the radii of the drums. It will be the product of the circumference of the drum into the number of coils of the cord.

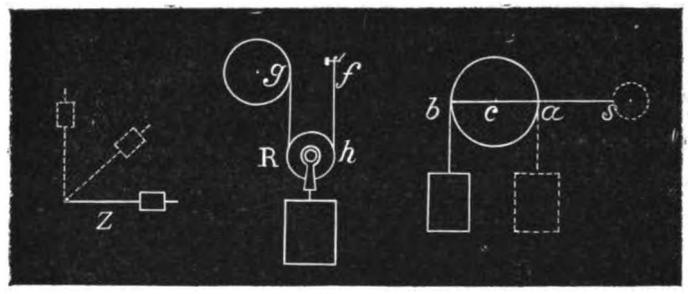

Fig. 67.

1193.—When the space available for the descent is very small the weight is supported by a movable pulley R (fig. 67), the groove of which rests on a cord attached by one extremity to the drum and by the other to a fixed support *f*. The duration of the fall is thus doubled, since the portion of the cord that is unrolled from the drum is divided equally between *g* R and *f h*; but the weight must be rather more than doubled, since the force exerted by it is divided, one part being supported by *g* and the other equal part by *f*; moreover the resistance occasioned by the pivots of R and the stiffness of the cord have to be overcome. These resistances become very serious when two or more pulleys are employed. (See the Theory of pulley-blocks and differential motion in any treatise on Mechanics.)

1194.—In winding up a watch or spring timepiece the spring is usually pulled in the direction of its action, so that the mechanism continues to go throughout the operation; but such is not the case with the ordinary form of weight clocks; for the winding stops the action of the motive power. This inconvenience can be remedied by using an endless cord, carrying a weight and counter-weight, or by adapting to the clock an auxiliary mechanism known as a Maintaining power. These

arrangements will be found described in most works on Horology.

System adopted by Julien Le Roy.*

1195.—This is equally applicable whether we employ a weight or a spring and fusee, and consists in bringing the point at which the impulse is applied between the axes of the two first mobiles; at *a* for example (fig. 67).

The relative forces exerted by the weight at *c* and *s* will be in the inverse proportion of the two levers *a c*, *a s*. It follows that the axis *c*, which would, if the weight were at *b*, support its entire pressure, will now only carry a portion of that pressure, the remaining portion being added to the force exerted at *s*. It will of course be observed that in this case the wheels turn in opposite directions.

SPRING MOTOR.

It cannot secure a constant force.

1196.—The motive force due to the tension of a spring is more or less variable. The causes of this want of uniformity are as follows:

(1) The elastic reaction of a spring becomes greater as the spring is further wound up;

(2) A metallic blade is very rarely homogeneous and worked with sufficient care to avoid different parts being of variable strength;

(3) Its energy alters with time, dependent on the duration and intensity of the flexure, and this change nearly always occurs irregularly throughout its length;

(4) Its elastic force diminishes slightly on elevating the temperature;

(5) Lastly a spring rubs against the bottom and lid of the barrel in uncoiling. The successive coils also adhere and rub together, either permanently or occasionally. All these resistances are from the nature of the case variable.

* A justly celebrated French horologist, who made the first turret clock with the mechanism placed horizontally. He was very accomplished and possessed extraordinary practical skill, which led him to introduce improvements in nearly all the branches of Horology of his day. On his death in 1759 he left four sons, all of whom became celebrated in different spheres of life; but the most noted was Pierre Le Roy, his eldest son and successor.

RELATIVE STRENGTHS OF MAINSPRINGS.

1197.—With a given thickness the power varies as the width, assuming all other conditions to remain the same.

1198.—For the same width and maintaining all other dimensions, such as length, diameter of barrel arbor, etc., *proportional*, the power varies with the square of the thickness (**1217**).

1199.—When the width and thickness vary, everything else, such as length, barrel arbor, etc., remaining proportional, the power will be as the width of each spring multiplied by the square of its thickness.

Observation.—It is clearly understood that the form should remain the same, as well as everything else, such as the material; for all steels have not the same elasticity, and the same steel may offer very different resistance according to the manner in which it has been worked, hammered, hardened, etc.

ON THE FORMS OF MAINSPRINGS.

1200.—With a view to diminish the differences in the pull of a spring when wound up to varying degrees and to increase its energy when nearly down, the central coils are not unfrequently made thicker. These springs, which are known in the trade as *cylindrical springs*, are characterized by the central turns when the spring is wound up rubbing with some force against each other, so that a portion of the motive force, which is excessive when the spring is fully wound up, is neutralized; and thus the difference in the power exerted by the spring in its varying positions becomes less. The advantage is, however, more apparent than real, since it depends on variable frictions that are modified still further by time, and a less thickness and better form of spring would exert a greater amount of useful force.

There are two forms of spring that possess real advantages:

1201.—(1) Those termed *tapered* springs, from the fact that the thickness of the band gradually diminishes throughout its entire length, commencing at the outer extremity. The effect of this form is to make the coils of the fully wound up spring separate, immediately the barrel begins to move, and on this account the spring is said to develope freely. If we compare this spring with the one that increases in the opposite direction, which we have referred to above, or to the spring of equal thick-

ness throughout, we find a greater variation in the force exerted as the spring runs down. This fact is in no way inconvenient in the case of chronometers, where a fusee is employed, and there is very great advantage in the more regular development of the spring and the reduction of disturbing friction between the coils.

1202.—(2) The form in which the thickness is the same throughout. With it the development is less uniform than with the spring just discussed, as is also the separation of the coils; but it has a counteracting advantage as regards ordinary watches in being more easily made and in exerting a less variable force, while, if the friction of the coils against one another is somewhat greater than with the chronometer spring, it is less than with the cylindrical spring; the variations, moreover, do not exceed the limits that ordinary escapements can neutralize, especially if the length of spring is well proportioned and the pivoted brace and stopwork are judiciously applied (**1188**).

Observation.—Of course the preceding considerations have reference to springs that are well made, of good steel, uniformly hardened and therefore of equal elasticity throughout. Otherwise, if the material and workmanship be bad, it is impossible to count on any regularity either in the present or future as regards friction, variations in the force, loss of elasticity, distortion of the spring, etc.

GENERAL OBSERVATIONS ON MAINSPRINGS,

Deduced from theory and experience.

1203.—Rounding of the edge of a spring; making its thickness slightly greater at the middle than the edges; cutting shallow radiating channels in the bottom and cover of the barrel, are all advantageous precautions. They diminish the friction surfaces, and therefore the resistance due to oil and adhesion (**36**).

1204.—Other things being equal it is better that the development of the spring be rapid rather than slow; for, as M. H. Robert has observed, the inconvenience known as the clustering of the spring diminishes as we increase the angular velocity of the barrel.

1205.—If there is any fear of variations in the force, principally on account of alterations in the oil and in the stickiness of the coils, the spring should be strong rather than weak. We would observe that, if, for example, the thickness

of a spring be doubled, its friction surfaces are very slightly increased whereas its motive force is quadrupled.

When a powerful spring is employed, it is especially important that the barrel and centre pinion depth, where each tooth remains in action for a considerable period, be made with great care, that the entire width of the ogive is in contact with the leaf, and, lastly, that this width of the face be sufficiently great (**38** and **40**).

1206.—The molecular strain of a mainspring is proportional to the time of its uncoiling and to the ratio that subsists between the thickness of spring and the diameter of barrel arbor. M. Rozé has determined by numerous experiments the limits in this proportion between which breaking or permanent distortion of the steel are generally avoidable (**1216**).

He has drawn attention to the very curious fact that when the barrel arbor is too small the molecular strain of the coils that are first wound up shows itself by a bulging outwards of the under side of the spring's internal extremity, while the outer face bends inwards in the direction of its length; this effect may sometimes be observed throughout a third of the entire length. It would be interesting to know to what degree this distortion, or variations in its amount, alters the elastic force of the spring.

TO STUDY THE DEVELOPMENT OF A MAINSPRING.

MEASUREMENT OF ITS FORCE.

1207.—The following is the most simple mode of studying the development of a spring:

Take a barrel fitted with a spring and arbor and with a cover cut away in the form of a cross in order that the action of the spring may be observed; then fix the extremity of the arbor in the jaws of a vice. Round the cylindrical face of the barrel fix a cord, which must be coiled as many times as correspond to the full winding up of the spring, and to the extremity of the cord attach a small balance-pan of known weight.

Successive loads are now applied (always taking into account the weight of pan) so as to balance the tension at equal intervals throughout each revolution; for example, each time give a quarter of a turn and examine the relative positions of the coils. The series of weights gives the ascending scale of

the motive force, and frequent observations of the positions of the coils will show the mode in which they are displaced: whether this occurs uniformly or with a strain towards one side; the manner in which the coils rub against each other, etc. The converse operation, that is the gradual diminution of the weight, will show the opening out of the coils and their separation from one another.

1208.—The energy of a mainspring can also be tested by means of the adjusting rod for fusees, simply holding the barrel in the hand or, preferably, holding it between the jaws of a watch-holder; and this is the method most frequently adopted. It seems useless to say more on the subject, as the instrument is known to all watchmakers.

Graphic representation of the variations in the power of a mainspring.

1209.—This is effected by employing the method of co-ordinates.*

The following explanation is intended for watchmakers that have no knowledge of geometry:

It is required to determine the law of the progressive increase in the force of a spring for successive angular displacements, each of 1/4th turn of the barrel (which may be represented by any pre-determined length in millimetres, etc.).

Draw the straight line A B and divide it into equal parts 1, 2, 3, 4, etc., each corresponding to 1/4th turn of the barrel, in other words equal displacements of the point of application of the force.

At 1, 2, 3, 4, etc., draw perpendiculars or ordinates the heights of which increase in the same proportion as the power, measured in, say, kilogrammes and fractions; we then have the perpendiculars 1 *c*, 2 *d*, 3 *h*, 4 *s*, etc.; and by drawing a line from the point *o* where the force is nothing through the highest points of these lines, we obtain the curve *o c d h s*, etc., representing the progressive increase in the force exerted by the spring. The abscissæ *o* 1, *o* 2, *o* 3, etc. are equal to *c' c*, *d' d*, *h' h*, etc.

1210.—The surface B *o h* R will represent the *mechanical work* (**124**) performed by the spring. This quantity, however,

* In this method, which is used in analytical geometry to determine the position of any point N (fig. 68, page 666) on a plane, the lines X *a*, Z *a*, are termed the *axes* of X and Z, *a b* is the *abscissa* and *b* N the *ordinate* of N, and the two lines taken together are called its *co-ordinates*.

has no importance as regards the springs employed in watch trains, but it has many other applications.

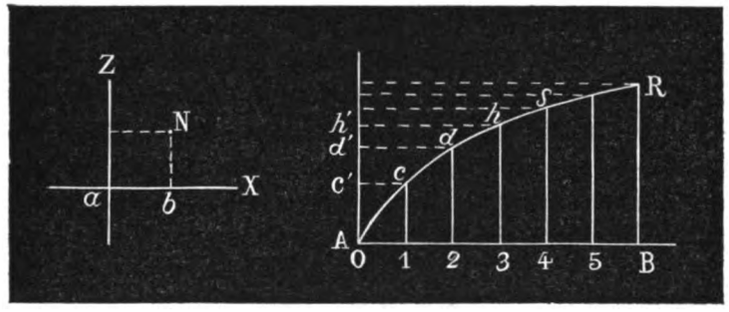

Fig. 68.

The drawing of such a figure is facilitated by employing paper that is already divided into sections by faint lines, as seen in figs. 1, 2, etc. of plate XVI. Assume that the two quantities taken to represent unity of force and unity of space traversed are as 2 is to 3, the paper must be divided in the proportion of 3, 6, 9, 12, etc., in one direction, and 2, 4, 6, etc., in the other; it will then not be necessary to measure the lengths of the co-ordinates.

The examples given in **1223** and the following articles will familiarize the watchmaker with this method.

MEMOIR ON MAINSPRINGS.

By MM. Rozé.

1211.—The earliest important work on mainsprings was published by MM. Rozé (father and son), in the *Revue Chronométrique*, volume II., pages 136 to 166.

More recently M. Résal, a distinguished engineer, has written a memoir on this subject, treating it in a different manner but without adducing any novel practical facts. We will then confine ourselves to the work of MM. Rozé, but, since the plan of this treatise does not admit of our reproducing it in its entirety, we will proceed to summarize it.

This memoir lays down and demonstrates the following theorems:

1212.—(1) *A mainspring in the act of uncoiling in its barrel, always gives a number of turns equal to the difference between the number of coils in the up and down positions.*

Thus if 17 be the number of coils when the spring is run down, and 25 the number when against the arbor, the number

of turns in the uncoiling will be 8, or the difference between 17 and 25.

1213.—(2) *With a given barrel, spring and arbor, in order that the number of turns may be a maximum it is necessary that the length of the spring be such that the occupied part of the barrel (exclusive of that filled by the arbor) be equal to the unoccupied part; in other words, the surface covered by the spring when up or down must be equal to the uncovered surface of the barrel bottom.*

Fig. 69.

Let *b b* (fig. 69) be the internal diameter of a barrel, *c c* the diameter of the arbor, *d* a circumference bisecting the surface between the arbor and edge into two equal zones A and B; the spring which will give the greatest number of available turns in this barrel will be such an one that its last coil when up and first coil when down coincide with the circumference *d*.

1214.—By making a small opening in the cover of the barrel it is easy to ascertain whether the spring gives the greatest number of available turns, for it is then possible to determine whether the external coil arrives at the point occupied by the internal coil when the spring is down. Article **1222**, moreover, gives another method whereby we can practically find without calculation the position of the circumference that gives equal surfaces.

1215.—Having determined this point and observing that when the spring covers a greater surface it is too long, and *vice versâ*, it becomes easy to find approximately:—(1) what would be gained or lost by shortening a given spring;—(2) whether there will be an increase or decrease in the time of going if a spring is replaced by another that is thicker or thinner.

1216.—*Diameter of the arbor.*—If the arbor is too large part of the elastic reaction of the spring will be wasted.

If too small there will be a rupture or straining of the spring and therefore a loss of elastic reaction.

It is, then, the thickness of spring that determines the diameter of the arbor or conversely, and from this it follows that the diameter is not an arbitrary quantity, since it depends on the duration of flexure and thickness of spring; this thickness can only be determined experimentally, depending as it does on the degree of flexure that can be applied to the spring without incurring permanent distortion. The flexibility of a steel spring is well known to be much greater when it is thin, but it varies somewhat with the quality of steel.

From experiments made by M. Rozé it appears that we shall not go too near the limit of elasticity of steel if the thickness of the spring is to the diameter of the arbor as 1 is to 26 or 34, according as the rotation of the barrel takes place more or less rapidly. Thus 1 : 26 is best suited to watches; 1 : 30 to chronometers; and 1 : 34 to clocks or timepieces that are intended to go several days.

Extracts from the Memoir.

1217.—"Remembering that the expansion as well as the contraction of a spring is directly proportional to the force applied, we find that, for the same width, the energy of a spring varies directly as the square of its thickness (**1198**).

1218.—"In the motors employed in ordinary horological mechanism, frictions occur that render the power very variable and make any experiments on this question liable to error.

1219.—"The power exerted in such motors is not the same when the spring is being coiled up as when it is going down.

1220.—The *mechanical work* performed by a spring varies inversely with the number of turns in its uncoiling (**124**)."

1221.—To these quotations we would add the following; "the length of a spring should never be altered until the conditions under which it is placed have been carefully studied;" and we will complete this summary of the memoir of MM. Rozé by describing their compass for determining the equal surfaces of the barrel.

Compass for determining the space to be occupied by the spring in a barrel.

1222.—This instrument consists of two movable right angled bars placed symmetrically opposite to each other, *c* and *d*, or *i* and *j* (fig. 7, plate XVI.). They form a square, and can

be made to slide over each other in the direction of the diagonal *c d* by means of an X-shaped piece formed of two jaws *m s, n r*. These are movable about a centre *v* between the two plates that carry the squares, and the plates are fixed to the jaws of the X by screws at *m, r, n, s*.

To use the instrument apply the diagonal *c d* to the space between the rim of the barrel and the arbor, and make the plates slide over each other by pressing against the projection *g* until *c d* is equal to this interval. The two extremities *o* and *o'* of the other diagonal are then points on the circumference that divides the space between the barrel rim and arbor into equal parts.

This instrument is based on certain geometrical properties of the rectangular isosceles triangle *j* in fig. 69, page 667.

Curves showing the variations in the force exerted by certain springs.

1223.—*M. Vissiere's chronometer springs.*—Fig. 1, plate XVI., is the curve for a spring that is controlled by a fusee; these are its elements:

Thickness at the outer extremity: at the middle of its height, 0·30 mm.; at the edge, 0·28 mm.

Thickness at the inner extremity: at the middle of its height, 0·25 mm.; at the edge, 0·23 mm.

Length, 1·30 metres. Height, 11 mm.

Internal diameter of barrel, 29·5 mm. Diameter of arbor, 9·2 mm. Turns in the uncoiling, 7·5. Mechanical work performed, 1·4 kilogrammetres (or 10·1 foot-pounds).

In fig. 2 plate XVI., the lower curve is also that of a fusee spring whose elements are:

Thickness: at inner extremity, 0·20 mm.; at outer extremity, 0·23 mm.

Length, 1·20 metres. Height, 11·5 mm. Internal diameter of barrel, 26·5 mm. Diameter of arbor, 8·7 mm.

Turns in uncoiling, 7·3. Work, 1·02 kilogrammetres (7·4 foot-pounds).

This same spring, after being shortened so as only to occupy half of the entire space of the barrel, gave the upper curve * * in figure 2, and was modified as follows:

Length, 0·900 metre. Turns, 7·66. Work, 1·185 km. (8·6 foot-pounds).

1224.—Fig. 3, plate XVI., is the curve given by a going barrel spring.

Thickness, 0·29 mm. Length, 1338 mm. Height, 11 mm. Diameter of barrel, 33·1 mm. Diameter of arbor, 8·7 mm. Turns, 10·25, spring occupying half the available space. Work, 1·88 kilogrammetres (13·6 foot-pounds).

The curve shown in fig. 11 of the same plate is given by a spring of the following dimensions: Thickness, 0·27 mm. Length, 960 mm. Height, 8 mm. Diameter of barrel, 26·4 mm. Diameter of arbor, 8·2 mm. Turns, 7. Work, 0·8873 km. (5·4 foot-pounds).

1225.—*M. Gontard's spring for a watch going* 15 *days.*—This spring, whose curve is shown side by side with that of a lever watch spring (fig. 12, plate XVI.), the two watches being of the same dimensions, had the following elements:

Thickness, 0·40 mm. Length, 1022 mm. Height, 3·8 mm. Diameter of barrel, 37 mm. Diameter of arbor, 12·6 mm. Turns, 7·12. Work, 0·60 km. (4·3 foot-pounds).

1226.—*Lever watch spring.*—Thickness, 0·22 mm. Length, 560 mm. Height, 1·55 mm. Diameter of barrel, 17 mm. Diameter of arbor, 5·4 mm. Turns, 5·5. Work, 0·0718 km. (·52 foot-pounds.)

1227.—*Ordinary clock springs.*—Fig. 13, plate XVI., shows the curve common to two springs in a clock that went for 45 days, with a pendulum weighing 1·5 kilo. traversing a supplementary arc of 15°; besides the striking with a double hammers.

SPRINGS THAT EXERT A SUFFICIENTLY UNIFORM MOTIVE FORCE.

1228.—By using a large barrel and a long spring of convenient form, of which only a few coils, selected by trial, are ever called into action, it is easy to obtain a motive force that varies very little between the two extreme limits to which the action extends. A. Breguet and U. Jurgensen have made marine chronometers with going barrels and springs of this description. It is, however, difficult to apply them to pocket chronometers because the spring, for want of space, cannot be of the requisite length and strength.

It is essential that great care be exercised in ascertaining the progressive force and in the construction of such a spring, in order that it may give a uniform pull and always keep the several coils apart, for otherwise it is impossible to avoid resist-

ance caused by adhesion and clustering. The spring used by Jurgensen in one of his chronometers without a fusee, which had a remarkably good rate, was very weak and had a length of about 3·5 metres. Its price was very high, and the maker did not care to make others at even that price.

1229.—M. H. Robert, who pointed out the objections to very long springs, has obtained a sufficient approximation to uniformity in the force of a spring of 10 or 11 turns in his going barrel chronometers; he only utilized 3·5 turns from the point of winding up, when the last turn was just out of contact with the barrel rim. It should be mentioned that his barrels have an angular velocity greater than that of the barrels ordinarily met with.

1230.—We do not know who it was that proposed to obtain uniformity in the motive force by subdividing the height of the barrel into two parts, and employing two superposed springs so arranged as to give a sort of equilibrium in the force. Nor shall we stop to discuss the value of this idea, which is certainly original, but the difficulty of its application is obvious.

1231.—A young Paris watchmaker, Viel-Robin, after many attempts to secure a uniformity in the force exerted by a mainspring, proposed the following arrangement: on one face of the barrel he fixed an elastic arm of a certain definite curvature (*a b*, fig. 8, plate XVII.). Its extremity entered an opening *b c* in the barrel, so as to be attached to the outer end of the mainspring. The two springs, the internal and external, were thus firmly united and resembled on a large scale the bent or Breguet balance-spring (*spiral coudé*) in use at the present day. When the mainspring was wound up the point of junction approached the centre in a manner and direction that were determined by the curvature and elastic force of the upper spring (*Revue Chronométrique*, vols. IV. and V.).

The author has assured us that the motive force is absolutely uniform, but we have had no opportunity of verifying this fact. Even ignoring the increased complexity of the barrel, it is to be observed that if this uniformity exists it cannot be maintained, except under the condition of its being made independent of the adhesion and friction of the coils, which is by no means an easy matter; and if friction or adhesion does occur, the conditions cannot be maintained invariable.

FREE MAINSPRINGS OF M. A. PHILIPPE.

1232.—For the last fifty years horological appliances that indicate and strike the hour, or repeaters, have been made in which one of the trains received its impulse from the motor direct, and the other was impelled by a barrel caused to revolve by a spring hooked on to the arbor and pressing against the rim; this spring being wound up either continuously or intermittently by one of the mobiles of the train that was acted on by the prime mover. More recently this method of driving has been applied to seconds trains.

Starting from this incomplete idea, M. A. Philippe proposed to bend the spring slightly backwards where it is attached to the barrel rim and to form in the inside of this rim four grooves in which the hook so formed on the spring would hold. It is retained in them with sufficient firmness to allow of the spring being wound up; and, when an attempt is made to wind beyond this point, the hook of the spring slips out of one groove and into the next, where it is arrested with a slight noise indicating that the spring is fully wound up.

M. Philippe then placed an elastic ring *a b c* (fig. 4, plate XVII.) in the inside of the barrel and attached the spring to its extremity. The spring was wound up in the direction indicated by the dotted line and arrow and was coiled up on itself so long as the elastic ring opposed a sufficient resistance; but beyond this point there was a slight displacement of the ring.

But the inventor did not stop here; he very soon saw that, in order to secure the advantages of these earlier arrangements under still better conditions, it was only necessary to strengthen the outer coil of an ordinary spring, and that he thereby obtained a new spring possessed of important properties.

It is in part represented in fig. 2 (plate XVII.). The portion between A and B has about double the usual thickness of the metal as seen at *f*. From B to about *c* the diminution is gradual, and then from *c* to *d*, *f*, etc., to the internal extremity that is attached to the arbor the thickness remains the same.

1233.—In a report on this subject to the Horological Society by M. H. Robert, we read:

"The stopwork becomes useless and may be suppressed. . . . . We draw attention to this fact without entering into the question as to whether it is of importance to the manufacturer or not to effect such an economy.

"This omission enables us to increase the height of the spring by the space formerly occupied by the stopwork.

"In repairing watches the free mainspring will be of great service when there is a difficulty in making the stopwork act with certainty in consequence of want of space.

"The most important point about M. Philippe's springs consists in the fact that they will replace the mainsprings with pivoted brace and, like them, will develope better than an ordinary spring." (If further information be desired consult the complete report in vol. v. of the *Revue Chronométrique.*)

In our judgment the detached spring, when used like an ordinary spring and doing away, as it does, with the necessity of the pivoted brace, has important advantages in helping us to attain to regularity. But without attempting to fix precise limits to its applications in the future, we are pleased to recognize in it an important step in advance in the horological art.

ON STOPWORK.—PIVOTED BRACE OR CURB.

GENERAL CONSIDERATIONS RELATING TO STOP MECHANISMS.

1234.—In common clock and watchwork it is a matter of indifference whether there is a variation in the rate of a few minutes in two or three days or between summer and winter; hence it does not matter whether they have stopworks or not, because it is almost useless to attempt to secure a satisfactory uncoiling with cheap springs, and in watches which, if they do give a fairly good rate, do so because a number of errors happen to balance each other and not as a consequence of the skill of the maker. It is logical, moreover, that if he finds it worth his while to effect a small economy in the price of the spring, he will save the sum that he should spend in providing a *good* stopwork. Thus watch repairers are compelled to suppress at least three-quarters of those that are supplied to watches, as they are badly planned, made, and adjusted; but the fault lies with the maker and not with the mechanism itself.

Description of various stopworks.

1235.—The most commonly employed is that known as the *Maltese cross* or *Geneva* stopwork shown at H, fig. 1, plate XVII. It is too well known to require any description.

1236.—Fig. 6, plate XVII. represents a modification of it that was suggested by M. H. Robert for use in his going barrel

chronometers. The stop takes place tangentially at *v* against the head of a screw that projects from the flat of the star-piece.

1237.—The arrangement shown in fig. 7, is due to M. Racapé; it is founded on the same principle as M. Robert's stopwork, the arresting of the motion being tangential. This form occupies less space than that of M. Robert, but the stop finger is somewhat less rigid.

1238.—Fig. 12, plate XVII., is the section of a barrel provided with the Gontard stopwork, named after its inventor.

The arbor *a b* has only one square at *b*, and thus the pivot *a* is longer than usual. On the face of the arbor nut *c* is left a projecting ring *s* (figs. 11 and 12), against which the concave faces *j* of the star-piece teeth move. This ring is cut away at *z*, and a pin *t* is planted opposite the space so left; this acts the part of the ordinary finger-piece. A sink cut in the inside of the cover gives room for the projecting ring and pin, and another sink on the external face at *h* (fig. 12) receives the star-piece. These two sinks overlap each other as shown by the circles *d* and *f* (fig. 11) and thus leave an opening *n* between them.

As the portion within the projecting ring on the arbor is cut away, it is possible to ensure a very safe bearing for the pivot in the cover.

This form of stopwork is very solid and, as compared with the ordinary stop, has important advantages which any watchmaker will be able to detect at once. We would only direct attention to the two following points that have been urged against it. The small opening is liable to allow of the escape of oil or to admit dust. It is impossible to set up the spring by one-quarter, one-half, or three-quarters of a turn except by rotating the cover in its sink, and it may probably not turn true.

The first objection does not appear to us very serious.

As regards the second M. Gontard states that in very many cases it would only be necessary to replace the hook of the arbor by one in the hole already drilled at the opposite side, but that in movements of his make the barrel covers always turn very true in the sink and are accurately fitted, so that there need be no difficulty in setting up the spring as required.

Action of the stopwork and pivoted brace.

1239.—In ordinary watches where it is hopeless attempting to secure a uniform development of a second-rate spring, the stopwork only serves to arrest the hand in winding.

With a spring whose maximum number of turns in winding up is considerable, the stopwork enables us to select and employ exclusively those turns that secure the most uniform action.

The addition of a brace has the effect of retaining a part of the outer coil always in contact with the barrel rim as it is fixed at a point somewhat below the hook, thus securing that only the hardened portion of the spring ever comes into action, a very great advantage; at the same time it renders the employment of stopwork essential, not only in order to arrest the hand, but also to prevent the spring from pulling too violently on the brace at the end of winding; for this would weaken the mainspring at the point of flexure and would cause the coils to be pressed too firmly against one another.

Some experiments made by a Swiss watchmaker, M. Lindemann, have convinced him that with a suitably placed pivoted brace the pull of the spring becomes more regular.

1240.—When the hook is fixed to the spring itself, as at T and V (fig. 10, plate XVII.), a very old device, the brace is unnecessary because there is no occasion to soften the extremity of the mainspring, and the hook, if firmly riveted, will maintain it sufficiently close to the barrel rim. The stopwork should be so set as to arrest the hand in winding just before the point at which the pull on the spring would become too oblique to the rim and thus strain the hook or rather its rivet.

1241.—In concluding the study of the subject under consideration we would observe that the foregoing details are sufficient for any intelligent watchmaker, and should enable him to decide when there is any advantage in employing a pivoted brace, as well as to determine upon the exact limits within which the stopwork should confine the tension of the mainspring (**1234** and **1247**).

Various suggestions for suppressing the stopwork.

1242.—Boussard, of Toulouse, suggested the use of an elongated bent bar. Assume *r* and *r'* (fig. 3, plate XVII.) to be the first and second coils of the mainspring. The white interval between them is occupied by the bent bar, which is also shown in elevation at A. This is made gradually thinner towards its extremity and extends about half round the barrel, or considerably beyond the point shown in the figure; it has a projection *b*, similar to that of a brace, that passes into a hole

in the cover, and a pin *e* is firmly riveted to A, being hardened and tempered with it. This pin passes through both the external coil of the spring and the barrel rim, thus replacing the ordinary hook.

Although this elongated bar, which reduces the pull on the eye of the spring, was received with considerable favour when first introduced it has not since then been very successful; for in addition to the fact that it requires to be made with great care, there is some difficulty experienced in placing it in position, the elasticity of the second spring is added to that of the mainspring when this is wound up, and the coils are more or less cramped according to the force exerted in winding, etc.

1243.—In repeaters, for more than half a century past, the small fixed barrel of the striking mechanism has been cut through on one side to allow the spring, which is hooked on the outside of the barrel rim, to pass through. In this particular case such an arrangement is very judicious, and its advantages are correctly explained in the writings published in the early part of the present century. The spring acts precisely as though it were braced and provided with a stopwork, because the motion of the push-piece is not sufficient to fully wind up the spring. The length of spring was so chosen that it gave about $3\frac{1}{2}$ turns, and of these only that one was used that gave the most uniform strokes; thus the spring never actually pulls against the edge of the rim, which, as we have already indicated, answers the purpose of a brace, while it retains the coils apart or only just in contact; and these precautions are very necessary when a light spring is called upon to drive a complete train of wheels in which a great variety of resistances can at any time be opposed to it and so prevent the strokes of the repeater taking place at equal intervals.

The variation in the pull throughout the action of the spring is very slight in the case we have been considering, as only one turn is used, but such would not be the case if all the turns were allowed to act; when wound up the coils would be forced close together and the entire force of the hand would be exerted against the edge of the rim, etc.

1244.—M. Sandoz, senior, of Paris, exhibited before the Horological Society an arrangement of the mainspring that was based on those of M. Philippe but differed in that the movable ring rotated in an opposite direction. For assume

a b c (fig. 4, plate XVII.) to be the elastic ring to which the spring is hooked, passing in the direction from *c* to *d*. When this is fully wound up the ring will be slightly displaced from *c* towards *b*, but without noise.

Several of those present pointed out that if a stud were fixed in the barrel rim at *a*, the act of winding would cause the extremity *c* to bend inwards and when it pressed against the outer coil of the spring it would answer the purpose of a stopwork; but, to the best of our knowledge, no one has ever applied for a patent for the invention.

1245.—At about the same time a Belgian watchmaker, of Seraing, introduced a very similar arrangement. He merely turned the spring itself backwards as shown in fig. 9, plate XVII. The end of the bent portion rests against the stud *c* and, on winding up the watch, the bend moves inwards and rests against the coiled up spring.

1246.—Lastly there was exhibited at the 1867 Exhibition in the case belonging to the Director of the Horological School at Besançon, an arrangement that more or less represents the two last forms. In it the spring (fig. 5, plate XVII.) is hooked at *s* to the extremity of a short lever *s b* which may either be simply held against a stud *b* or movable about this point, having a short axis that traverses the cover and bottom of the barrel like a pivoted brace.

Concluding observations on stopworks and the uncoiling of springs.

1247.—We do not propose to examine into these various attempts critically; readers that have carefully studied the foregoing chapters can do this for themselves, and will be able at once to determine the practical value of each. Nevertheless it is well that we should repeat: the object of the stopwork and brace is to facilitate the selection of the most uniform turns of the spring, so that the variations shall not exceed what the escapement is competent to counteract, and a further object is to preserve as far as possible the elastic force of the mainspring, which, as we know from experience, is liable to diminish in parts or throughout the entire length if the spring is subjected to too great tension or flexure.

If the arbor is too small the stopwork will enable us, by employing a spring that is relatively long, to retain one or two turns of the spring always in contact with it, and thus, in effect, increase its diameter.

It is known that the ordinary springs of commerce are marked by very considerable pressure and friction between certain coils; and that the sudden separations that result, which vary according to the condition of the oil, interfere with the rate of the watch. It will in most cases be found that the force exerted is not only more uniform, but very often greater if we employ a spring that is properly braced and thinner, so that there are a greater number of turns in the uncoiling; the most uniform series of these will of course, through the action of the stopwork, be alone used. This may be considered paradoxical, but it is strictly true. When the uncoiling is less constrained there will always be a slight increase in the force.

The power of selecting the best series of turns to include within the limits of the stopwork is so important that it must be regarded as the reason for the retention of the fusee in chronometers; because with the fusee we can secure with certainty a rather longer period of going and a mainspring the uncoiling of which takes place under the best possible conditions (**1201**).

This last observation is of the highest importance, so much so that it alone should settle the discussion between the advocates of the fusee and of the going barrel. It explains why pocket chronometers, unless they have enormous barrels, cannot do without a fusee. It justifies the English makers who in their splendid high-class watches, with such excellent rates, have retained it; but they make an error when they put it in watches of the commoner class.

Conclusion.—In watches that are expected to possess a good rate, the first element of regularity consists in the proper uncoiling of the mainspring with a minimum of friction between the several coils.

This property of uncoiling satisfactorily depends upon: the form of spring, its mode of attachment, and the judicious use of a brace when necessary.

If we have a spring possessing the requisite properties, it will only retain them on condition that it is not subjected to strains in excess of the limits of its elasticity, and it is the stopwork that must guard against such an accident and prevent the condition of the spring being modified.

If we have only the common spring met with in ordinary watches, we can only repeat what is said in article **1234.**

PART III.

MISCELLANEOUS ARTICLES.

INTRODUCTION
TO THE STUDY OF ISOCHRONISM AND COMPENSATION

LAW OF ELASTICITY AND ISOCHRONISM OF VIBRATING LAMINÆ AND RODS.

1248.—The maximum elasticity of a body is determined by its limit of compression, or tension, or torsion, that is to say the point beyond which, if released, it will not regain its original form. It has in fact become distorted.

The charge that corresponds to this limit in delicate bodies is a small fraction of that which would determine its rupture, and this breaking strain gives a measure of the tenacity.

For a metallic wire or lamina the limits beyond which distortion would occur are the nearer together according as the piece of metal is shorter and thicker; and it is only between these limits that the body will exactly recover its original form so as to conform to the following laws, which assume the metal to be absolutely homogeneous.

Elasticity of flexure.

1249.—*Two laminæ or rods that only differ as regards length will, in a given time, perform a number of vibrations inversely proportional to the squares of their lengths.*

Length of $a\,b$ (fig. 70) $= 1$. $1 \times 1 = 1$

„ $a\,c$ $= 4$. $4 \times 4 = 16$

Hence $a\,b$ will perform 16 vibrations while $a\,c$ performs 1.

1250.—*For a given length, the number of vibrations of a rod or lamina is proportional to the thickness but independent of the width* (assuming the vibrations to take place entirely in the plane perpendicular to the broader face).

Thus let $f2$, $f4$, and $g\,4$ (fig. 70) be the sections of three rods; $f2$ and $f4$ will make the same number of vibrations, and $g\,4$ will make twice this number in a given time.

Elasticity of torsion.

1251.—*With a given stretching weight, the periods of oscillation of two rods or wires of the same length but different diameters are in inverse proportion to their sections.*

Take two wires *e* and *d* (fig. 70) whose diameters are 1 and 2.

Section of *d* is proportional to 1×1 or 1
„ *e* „ 2×2 or 4

The period of vibration of *d* will be to that of *e* as 4 : 1.

Thus for round rods or wires the period is inversely proportional to the square of the diameter.

Now take two rods, *f* and *g*, of equal length but square section.

Section of *f* = 2×2 or 4
„ *g* = 4×4 or 16

The proportion is then as 4 : 1 as in the case of round rods.

1252.—*With a given wire the numbers of oscillations in a given time are as the square roots of the stretching weights.*

Assuming the weights to be . . . 1, 4, 9, 16, 25...
These numbers will be . . . . 1, 2, 3, 4, 5...

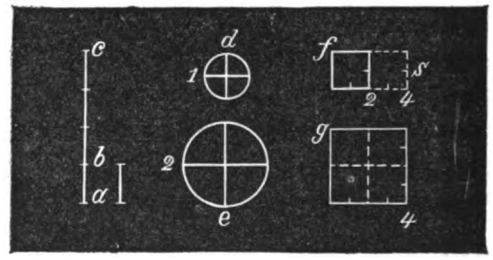

Fig. 70.

1253.—*If the weight remain the same and the length be varied the period will depend on the square root of the length.*

Thus for lengths. . . . . 1, 4, 9, 16, 25...
The corresponding periods are . . 1, 2, 3, 4, 5...

Observation.—These laws can be easily verified for the torsion of a lamina or wire, since it is only necessary to find from tables the weight and amount of displacement that are possible without exceeding the limits of elasticity, as gravity does not interfere with the motion of the weight.

But in the case of flexure, if the force exerted by the lamina is such that it can, without constraint, overcome the weight,

the force of gravity will be almost neutralized, whereas if the blade is influenced by the weight, its action is subject to the control of gravity; its elastic force goes for very little and the apparatus becomes nothing but a pendulum under the action of gravity and, as such, its oscillations become in a great measure independent of the weight, being a function of the distance between the centres of suspension and oscillation.

Law of the Isochronism of Vibrations of elastic laminæ.

1254.—The oscillations or vibrations of a metallic lamina or wire are only *isochronal*, that is to say performed in equal times, when they are *extremely short.*

Hooke, F. Berthoud, and certain physicists of their day, were in error when they concluded from some incomplete experiments, performed without any sufficient means of verification that, "*the vibrations, both long and short, of a given spring are isochronal.*"

1255.—When a rod or lamina carries a weight (which must not be sufficient to strain it) *the oscillations of varying extent will*, in virtue of a mechanical law, *be isochronal when the force or rather moment of torsion is proportional to the angle of torsion.*

This law, that the forces must be proportional to the angular displacement, forms the basis of isochronism (**661**).

1256.—Since unequal arcs of vibration can only be isochronal when performed in obedience to the above law, it follows that any appreciable alteration in the weight, which may increase or diminish the inertia and the pressure on pivots as well as change the resistance and therefore the conditions of the lift, may modify the isochronism for better or worse according to circumstances. For variation of the weight of a body supported on pivots and under the control of a spring will cause not only the pressures but also the resistances to vary; either during the lift when the motive force is being applied, or towards the end of each oscillation when the swing will possess all the more energy as the mass in movement is greater. If the relative proportions of the forces to the angular displacements are altered the isochronism will be modified in a similar degree.

These considerations enable us to explain why a balance-spring that is isochronal with the elastic balance (**1424**) is usually no longer so in the chronometer when it has to overcome the various resistances of the escapement.

ON COMPENSATION FOR THE EFFECTS OF TEMPERATURE IN MACHINES FOR THE MEASUREMENT OF TIME.

Law of Dilatation.

1257.—Bodies increase in volume with an elevation of temperature and diminish when it falls. The balance of a watch and the pendulum of a clock change their dimensions with every variation of temperature; and the same is the case with all other parts of the machine.

The elongation of a body in any one direction with heat is known as its *linear dilatation* or *expansion.*

The increase in its volume, that is in all three dimensions, is the *cubical dilatation;* this depends on its linear dilatation in length, breadth and thickness.

1258.—Experience has shown that the *dilatation of metals* and nearly all solids* is practically *uniform* from 0° to 100° C., in other words that it is proportional to the elevation of temperature.

Beyond these limits it becomes variable and seems as a rule to increase more rapidly as the temperature is elevated.

1259.—In a perfectly homogeneous metallic body that is free from constraint *dilatation takes place in straight lines in every direction from the centre towards its surface.*

A disc of cast steel that is homogeneous and accurately turned will expand uniformly, always remaining flat, so that its circumference will become larger but remain perfectly circular.

A disc of wood (as well as of some non-homogeneous metals) will become oval and may even warp on heating; this is owing to the fact that its linear dilatation is not the same along the fibres of wood as it is across them.

1260.—We would here insist on the exact enunciation of the law of dilatation as given above, because ignorance of its nature or careless observations of its effects have given rise to very erroneous ideas among watchmakers and even, what is more remarkable, among some physicists of repute, and have thus given rise to the invention of a host of ridiculous modes of compensation.

* Among the bodies that form an exception and do not expand with a rise of temperature may be mentioned water below 4° C., indiarubber, and clay.

As a general or rather invariable rule, *when the effect of dilatation is not in conformity with the law above enunciated* it is owing to the fact that the metal is non-homogeneous; that portions of it are crystalline or present flaws or breaks of continuity; that foreign bodies are unevenly distributed throughout its mass; that, owing to the torsion, hardening, rolling or coiling, either the whole or part of the body is in a state of molecular constraint, in other words of compulsory equilibrium which tends to disturb itself on the application of heat or cold, or under percussion, pressure, etc. This condition of unstable equilibrium is one cause of want of uniformity in the effects since they do not always remain identical in the backward and forward movements (**1267**).

A body that is unevenly heated or which, owing to differences in its molecular constitution, does not receive the heat uniformly will be the more strained according as its molecular state is less uniform (**351**).

Force of Dilatation and Contraction.

1261.—The *force of dilatation* of a body is equal to the resistance it is capable of opposing to compression, and can be represented by the weight which, for example, pressing vertically on a bar of iron would be competent to compress it by the exact amount that it expanded, when free, on an elevation of temperature of 1° C.

The *force of contraction* is equal to the resistance the body can oppose to traction, and is measured by the weight that would be competent to elongate the body to the same extent as it is shortened by a fall of temperature of 1° C.

COMPENSATORS.

General Considerations.

1262.—As we have seen, the dimensions of a balance or pendulum are always varying; for this reason attempts have been made to neutralize the effects of these changes on the rate of a timekeeper by what are known as compensation balances and pendulums.

Many makers of compensators that are more or less novel fall into an error in neglecting the sources of gain, etc., that

arise in the mechanism itself and they only attempt to neutralize the dilatation of the balance or pendulum. As a matter of fact this latter has but little influence in high-class horological mechanism, and, as regards ordinary timekeepers, it is nearly always very much less than the gain or loss with changes of temperature that arise from the machine as a whole and are often due to faults of construction.

Clocks and watches that are intended for ordinary use will, if made with care and well proportioned, do very well without compensation except in a few special cases.

But this is not the case with high-class mechanism that is intended for the measurement of time to within a few tenths of a second; in such a case compensation is indispensable.

Preliminary details relating to the Properties of metals.

1263.—Before attempting to make or plan any form of compensation, a watchmaker must have accurate notions as to the nature of metals, and he should study, both experimentally and in works on physics:

Their *dilatability* or elongation for each degree rise of temperature;

Their *conductibility* or the greater or less facility with which they receive and transmit heat;

Their *capacity* for heat, that is the power they possess of absorbing a quantity of heat, on which will also depend the greater or less time occupied in losing heat;

Their *density* or the relative weights of equal volumes (**47**).

It is further necessary to know their *elasticity* and the manner in which it is modified by temperature. It will then be easier to understand the effects of centrifugal force at different temperatures; for if the elasticity of the divided rim of a compensation balance diminishes or increases, centrifugal force will have a greater or less effect.

1264.—F. Berthoud concluded from his experiments that about eleven-twelfths of the effect of a compensator is required to counteract the loss of elastic force of the balance-spring.

In one of his marine chronometers he stated that the relative loss to be corrected was:

62″ of loss due to the balance itself;
331″ „ „ balance-spring.

Selection of Metals.—Concluding observations.

1265.—The first difficulty to be overcome in making a compensator consists in selecting the metals, which seldom possess the exact qualities that are required. And when we do obtain suitable metals it may happen that their properties are damaged in part or completely by bad workmanship.

Glass is not satisfactory; it does not expand uniformly.

Zinc is too crystalline and non-homogeneous except when extreme care is devoted to the melting and preliminary working.

Steel is liable to be magnetized and it is, moreover, crystalline.

Ordinary *brass* is very seldom homogeneous; very often it exhibits different proportions between the two metals in different parts of its mass, due to the phenomenon of *liquation*. After fusion and indeed when it has merely been roughly worked, it remains for some time in a condition of molecular re-arrangement. This has been more especially noticed in parts that are made of the cast brass of commerce.

An analogous effect has also been shown to occur in the case of steel laminæ when they have been subjected, under certain conditions, to long continued vibrations.

Iron and *copper* require less care in their preparation; but, while they are not entirely free from the objections mentioned above, they are softer and less rigid, the one than steel and the other than brass.

1266.—We trust that the reader will now understand the conditions under which we can hope to neutralize the effects due to changes in the dimensions of bodies, owing to variations of temperature, in machines for the measurement of time; and, in conclusion, we would repeat what we have already said in the *Revue Chronométrique*: "As a general rule, every compensation pendulum or balance whose construction is characterized by harsh and numerous frictions, many points of contact and excessive pressure supported by small surfaces, is bad, and if it does not prove itself to be so at once it will in time. Friction and numerous contacts against any one part will interfere with the effects of dilatation; small surfaces subjected to considerable pressure will wear or become distorted in time, to say nothing of the adhesion which is a further cause of variation."

1267.—As an example we would refer to the system of

compensation by levers, in which at certain periods the resistance to compression counteracts the force of dilatation owing to the great pressure and adhesion that occur. To establish this fact it is only necessary to shake the entire arrangement or to give it a slight sharp blow, when the dilatation will suddenly assert itself. It seems moreover to be established at the present day that dilatation occurring at points of contact takes place, as in the case of crystalline metals, by a succession of very slight movements or infinitesimal jumps; and it naturally follows that if the normal action is checked the dilatation will occur with all the more violence.

1268.—The preceding considerations will explain why the regularity attainable by employing a compensation balance with bi-metallic arms and weights that move freely on a change of temperature has never been surpassed by any other combination, in which a greater number of parts exist that are subject to several contacts and pressures. In fact there is always in such an arrangement a loss of time between the backward and forward movements, as well as slight differences in the non-uniform motion due to dilatation, which are insensible in large pendulums such as the gridiron, where the pressure is well distributed, but become very detrimental in the very delicate and easily affected balances of chronometers.

THE PENDULUM.

Historical Notice.

1269.—The invention of the pendulum as a means of measuring short intervals of time is attributed to Galileo. After his day astronomers counted the number of oscillations performed in a given time, and employed this datum to measure the periods of celestial phenomena.

Some authors maintain that the Arabian astronomers employed the pendulum, but there is no evidence to show that Galileo was aware of this.

If a slight angular movement be given to a freely suspended pendulum, its oscillations, while gradually diminishing in extent, will occupy periods which at first sight Galileo affirmed to be equal. But he was mistaken; the difference, although very slight with short arcs, is none the less real, and Huyghens dis-

covered and proved that the oscillations of a pendulum are only isochronal when its centre of oscillation describes a cycloidal path.

The first application of the pendulum to clocks is also due to Huyghens (1657); but it is only right to remember that Galileo had some idea of this adaptation, and that he had even invented an escapement with a view to carrying it out. His son, Vincent Galileo, either from not seeing the value of the discovery or from being engaged on matters of another kind, neither published it nor followed it up.

1270.—Compensation pendulums, whose object is to neutralize the effects of contraction and dilatation due to changes of temperature, are of various kinds.

The *mercury* pendulum was invented by Graham, who also suggested the *gridiron* subsequently brought into practical use by Harrison.* Regnault proposed the pendulum with compensating weights, and Ellicot and Deparcieux made the first lever compensations, of which very many forms exist at the present day owing to numerous watchmakers having directed their attention to pendulums of this class. Subsequently Varinge and Rivaz and others proposed to effect the compensation by employing an iron or copper tube containing a second tube of some more expansive metal, which is supported at its lower extremity, while within it is an iron rod fixed at its upper end and carrying the pendulum bob. The mode of action is identical with that of Harrison's gridiron pendulum.

THEORY OF THE PENDULUM.

1271.—A theoretical pendulum consists of a heavy molecule suspended at the extremity of a perfectly flexible cord without weight and oscillating in a vacuum.

Such an ideal pendulum could not of course exist, but it is realized as nearly as possible in practice by suspending a small platinum ball by a thread of silk; this is termed a *simple pendulum.* At the latitude of Greenwich a simple pendulum if displaced from the vertical and allowed to oscillate *in vacuo* will perform *one oscillation in a second* when its length, measured from

* A very clever English watchmaker, born in 1693 and died in 1776, who invented bi-metallic compensation and designed a marine chronometer that secured him a Government award of £20,000. His arrangement was once copied by Larcum Kendal, but afterwards abandoned, as the English preferred the form suggested by J. Arnold, which consisted in an ingenious application of the principles discovered by P. Le Roy.

the point of suspension to the centre of the small heavy ball is 994·13 mm. (39·1397 ins.). For an approximation when no special accuracy is required, as in the case of ordinary horological mechanism, we can take 994 mm. or 39·14 ins.

Laws of the movement of simple pendulums.

1272.—*The numbers of vibrations in a given time are inversely as the square roots of the lengths.*

If the bob is displaced from the vertical and released it will return and ascend to an equal distance on the other side in virtue of its weight. The velocity of movement of the pendulum is in accordance with the laws of falling bodies (**126**), for the moving pendulum is no more than a falling body under certain restrictions. If we assume the pendulum to be displaced laterally until its rod is in a horizontal position, it will be seen that the distance through which it descends is equal to the length of the pendulum (N and *n*, fig. 72, page 694). Hence it follows that the descent of a short pendulum will, in virtue of the laws above referred to, take place in much less time than that of a long one. Thus, consider the case of a short pendulum whose length (*n*) is a quarter of that of the longer one (N); the shorter will travel twice as quickly as the longer. Or, in other words, it will perform two oscillations while the longer performs one. The lengths are as 1 : 4 and the square roots of these numbers are 1 and 2; thus the number of oscillations are inversely as these square roots.

1273.—*The times occupied in the descent* (or the periods of the oscillations) *are proportional to the square roots of the lengths.*

If the longer pendulum fall in 2 seconds, the shorter falls twice as quickly and will therefore reach the vertical in 1 second; and 2 and 1 are the square roots of the lengths 4 and 1.

1274.—*The lengths are inversely proportional to the squares of the number of oscillations in a given time.*

If we observe that:

The lengths are	1 and 4
The corresponding numbers of oscillations .	2 and 1
The squares of these numbers	4 and 1

we have some evidence of the truth of this law.

These several laws will enable us to determine the length of pendulum for any case that presents itself.

To determine the length of a simple pendulum, the number of oscillations being given, or vice versâ.

First method.

1275.—Let the pendulum be required to perform 7,000 oscillations in an hour.

The simple seconds pendulum measures 994 mm. and it makes 60 × 60 or 3,600 oscillations per hour; we thus, from **1274**, have the proportion:

$$7{,}000 \times 7{,}000 : 3{,}600 \times 3{,}600 :: 994 : x$$
$$\text{or} \quad 49{,}000{,}000 : 12{,}960{,}000 :: 994 : x$$

Dividing the product of the means by the known extreme we obtain

$$x = \frac{12{,}960{,}000 \times 994}{49{,}000{,}000} = 262{\cdot}9 \text{ mm. } (10{\cdot}35 \text{ ins.})$$

1276.—If the length of a pendulum be given, say 121 mm. (4·764 ins.), the number of oscillations will be calculated in accordance with article **1272**:

$$\sqrt{121} : \sqrt{994} :: 3{,}600 : x$$
$$\text{or} \quad 11 : 31{\cdot}525 :: 3{,}600 : x$$

whence we obtain

$$x = \frac{31{\cdot}525 \times 3{,}600}{11} = 10{,}317;$$ the required number of oscillations.

It will thus be seen to be very easy to find the length of the pendulum for a given number of vibrations, or *vice versâ;* but, although such calculations are almost quite useless, since a table is given, at the end of the volume, of the lengths of pendulum for all numbers of oscillations, we have thought it best to give the above details for the benefit of readers that are unacquainted with the subject; in order that they may become more familiarized with it and see how confusing such calculations become if logarithm tables are not employed. However, in case such problems have to be solved, we will proceed to explain a second method that is rather more simple, which has been given by M. Millet in the *Revue Chronométrique.*

Second method.

1277.—Take as a basis for calculation the pendulum that performs *one oscillation in an hour,* the length of which is 12,880,337·93 metres (507,109,080 inches) or, in round numbers, 12,880,338 metres; by the third law (**1274**) we obtain the following proportion:

$$12{,}880{,}338 : x \text{ (the length)} :: v^2 \text{ (the velocity)} : \frac{1^2}{44}$$

Since the square of 1 is 1, it is only necessary to replace x by the length (if this is given), or v by the number of oscillations in an hour (if they are pre-determined), and the value of the unknown quantity will be obtained.

First Example.—How many oscillations will be made by a pendulum measuring 305 mm. (12·008 ins.)?

We have the proportion; 12,880,338 : 0·305 : : v^2 : 1

Dividing the product of the extremes by the known mean

$$v^2 = \frac{12,880,338}{0·305} = 42,230,616,$$

and v will be the square root of this number, or 6,498 oscillations per hour.

If the dimensions are given in English inches the numbers 507,109,080 and 12·008 would be employed thus

$$507,109,080 : 12·008 : : v^2 : 1$$

$$\therefore\ v^2 = \frac{507,109,080}{12·008} = 42,230,936,$$

the slight difference in the results being due to the non-equality of the two approximate figures given above.

Second Example.—What should be the length of a pendulum to give 4,100 oscillations per hour?

It will suffice to indicate the several stages of the calculation:

$$12,880,338 : x : : 4,100^2 : 1$$

$$12,880,338 : x : : 16,810,000 : 1$$

$$\therefore\ x = \frac{12,880,338}{16,810,000} = 0·766 \text{ metres } (30·158 \text{ ins.}).$$

The Isochronal Pendulum.

Scale of velocities.

1278.—As we have already seen, Galileo concluded from his observations, which must necessarily have been very imperfect with the means at his disposal, that the long and short arcs with any given pendulum are isochronal. Huyghens detected the error made by Galileo, and showed that the long arcs occupy a longer period than the short arcs, and he further demonstrated that, in order to make all the oscillations isochronal, the centre of oscillation of the pendulum must describe a cycloidal path on either side of the vertical. This cycloidal path gradually deviates (inwards) more and more from the circular arc described from the point of suspension as a centre,

assuming the lowest points of the two to coincide; and from this fact we see that the retardation of the long oscillations in a circular arc will become greater as their amplitude is increased.

This retardation is deducible from the laws of motion, by which it can be shown that if a ball roll along the concave face of a circular arc it will occupy a longer period in descending from *c* to B (fig. 71) than from *a* to B, etc., whereas if the ball roll along the internal face of a cycloid *f d* B, the spaces *d* B, *f* B, etc., will all be traversed in the same period of time.

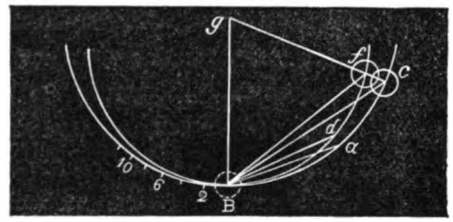

Fig. 71.

The velocity acquired at B by a pendulum that has traversed the circular arc *c* B is to that of the same pendulum when it commenced its movement at *a* as the chord *c* B is to the chord *a* B. Hence it follows that if we subdivide the arc traversed, commencing from B, into parts such that their chords are in the ratio of 1, 2, 3, 4, 5, etc., the arc so divided will give us a *scale of velocities;* so that, if the pendulum start in two cases from the divisions 10 and 5 for example, its velocities at B will be as 10 is to 5.

1279.—Huyghens proposed to cause the pendulum suspension, formed of silk threads or very flexible metallic blades, to bend by coming in contact with curved metallic cheeks so as to make the pendulum bob traverse a cycloidal arc. This method, although very ingenious, was not successful; the effects of moisture, dilatation, friction, adhesion of contacts, etc., gave rise to irregularities that were greater than those he desired to counteract.

Circular arcs were therefore reverted to and in the common clocks the loss on the long arcs was more or less balanced by a slight recoil of the escapement; as regards very accurate timekeepers, by devoting great care to their construction it was possible to reduce the extent of the oscillation to only a few degrees. In such a small interval the cycloidal and circular arcs are practically coincident, and the error committed in as-

suming these very small oscillations to be isochronal was exceedingly minute; but makers went to an extreme: they replaced long cycloidal arcs performed by comparatively light pendulums by pendulums of immense weight with arcs of oscillation that were almost imperceptible. Then it only required a moderate shake to stop the movement.

We have already indicated in the articles on clocks and timepieces commencing at page 541, the limits between which experience has shown that the angular movements of pendulums should be confined, and we would therefore refer the reader to them.

The retardation of a pendulum caused by an increase in its arc of oscillation.

1280.—The retardation caused by increasing the arc of oscillation of an entirely detached pendulum increases in a much more rapid ratio than the amplitude, as will be seen from the following table which is due to M. H. Robert.

Arc of Oscillation.	Loss in 24 hours.
0·5°	0·43 sec.
1·0°	1·55 „
2·0°	6·60 „
3·0°	14·80 „
4·0°	26·25 „
5·0°	41·30 „
6·0°	1·05 min.
7·0°	1·15 „
8·0°	1·35 „
9·0°	2·15 „
10·0°	2·50 „
15·0°	5·60 „
20·0°	10·70 „
25·0°	17·00 „
30·0°	24·00 „

A change of latitude or of altitude will occasion a loss or gain in the rate of a pendulum.

1281.—Gravity, or the mutual attraction between bodies, varies directly with their masses and inversely as the square of their distances. From this it follows that the farther a pendulum is removed from the centre of the earth the less will it be attracted in its descent towards the vertical. This is one reason why a pendulum loses on being transferred from a plain to the top of a mountain or from one of the earth's poles towards its equator, for the earth is not a perfect sphere but a spheroid slightly flattened at the poles.

Another reason why the pendulum loses on being transferred to the equator lies in the fact that the rotation of the earth gives rise to centrifugal force at its surface; from being zero at the poles this force gradually increases to a maximum at the equator, and, since it acts in opposition to the force of gravity, it counteracts a gradually increasing proportion of this force.

A pendulum that beats seconds at Greenwich would require to be shortened rather more than 3 millimetres if transferred to the equator, and lengthened 2 millimetres on being removed to the pole, in order that it might continue to beat seconds.

THE ORDINARY OR COMPOUND PENDULUM.

To find its centre of oscillation.

1282.—We have hitherto only been considering the simple pendulum as defined in article **1271**, whereas the pendulums employed in practice cannot dispense with a sufficiently rigid rod, which must possess weight; it is, moreover, often necessary to attach pieces to it. Thus the centre of oscillation will be caused to rise above the centre of the bob by an amount that increases as the weight of the rod is made greater. As many watchmakers have a difficulty in distinguishing between these two centres we will consider them in some detail.

Consider the case of an oscillating pendulum (fig. 72, page 694). In virtue of the laws of gravity as influencing a pendulum, a has a tendency to move quicker than b, b quicker than c, and so on; and thus it follows that, whereas the portion d tends to retard the movement, the parts a, b, c, tend to accelerate it. There must be some point, say x, at which these two tendencies neutralize each other, so that the motion will be the same as though the entire mass of the pendulum were concentrated at that point, *which is known as the centre of oscillation;* and the oscillations of the compound pendulum will be isochronal with those of a simple pendulum of length $y\ x$. These effects may be rendered visible by using a pendulum formed of some flexible substance carrying two bobs at different heights. The acceleration of the upper bob will be evidenced by the flexure of the rod.

Hence the heavier a pendulum rod is made or the more it is loaded with accessory pieces, the higher will its centre of oscillation be above the centre of the bob; and a compound pendulum of whatever form or material has exactly the same

period of oscillation as a simple pendulum whose length is equal to the *distance between the centre of suspension, y, and the centre of oscillation, x*, of the compound pendulum (neglecting differences due to the resistance of air, the suspension spring, etc.). It will be seen then that attention must be directed to the determination of these two points and not to the external dimensions of the pendulum.

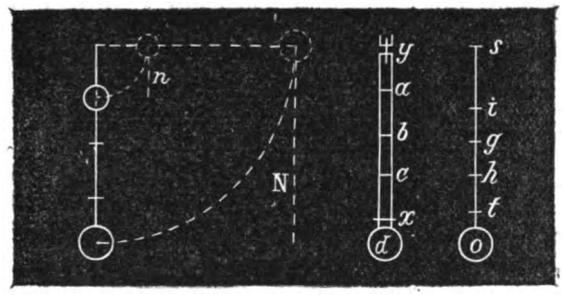

Fig. 72.

In a pendulum formed of a heavy bob and a light rod of aluminium for example, the centre of oscillation will be very slightly above the centre of the bob, and it may even be at it or below it if the screw that supports the bob counteracts the weight of the rod.

In a seconds pendulum with a lenticular bob weighing 10 kilogr. supported by a steel rod 11·33 mm. wide and 4·5 mm. thick, the centre of oscillation will be about 8 mm. above the centre of the bob.

1283.—It will be seen that in questions relating to the pendulum no certain measurements can be obtained unless the simple pendulum is taken as a starting point. So that any observations made on this latter will be applicable to the compound pendulum.

In ordinary practice the method explained in article **1016** is adopted for its determination; but by such a means only an approximation is obtained, because it in reality only gives the centre of gravity (**48**), and the centre of oscillation will be at some distance from the centre of gravity, especially if the pendulum rod is loaded at its upper end, by a knife-edge for example; or if it is provided with attached pieces such as a slide (see article **1305**), if the suspension spring is of some length, etc.

The theory of the pendulum, one of the great researches of Huyghens, is complicated and can only be studied by mathematicians; we must then confine our attention to those points.

with which the watchmaker must be acquainted, and certain experimental data. In any case if a watchmaker requires to find the centre of oscillation of a pendulum he can always resort to a very simple method: having suspended a small ball of platinum by a very fine light thread, hold the thread so that the distance between the hand and the centre of the ball is equal to the length given in the tables for the required number of oscillations. By making this simple pendulum and the one under examination oscillate one in front of the other, it will be easy to fix upon the length of the latter that gives oscillations that are isochronal with those of the former and we shall further know, very approximately, its centre of oscillation.

We can also practically determine the position of the centre of oscillation as compared with the centre of rotation (when this is a knife-edge) by the principle, which can be proved mathematically, that they are convertible. That is to say, if the pendulum is inverted so that it can be supported at its centre of oscillation and is caused to oscillate, the period of an oscillation will be the same as when it is supported at the centre of suspension.

To make such an experiment a small knife-edge is usually attached just above the centre of the lenticular bob of the pendulum in such a manner that it can be moved upwards or downwards by means of a screw. The pendulum is then inverted and made to oscillate on this new centre, which is raised or lowered until the pendulum is found to perform precisely the same number of oscillations in a given time in each of the two positions. The distance between the knife-edges will then give the length of the corresponding simple pendulum.

When the suspension is by a spring or thread this method is not sufficiently exact.

On the form of a pendulum.

Resistance of the air.—Barometric Pressure.—Berthoud's experiments.

1284.—The maximum regulating power of a pendulum with a given motive force is secured by concentrating the greatest possible amount of its total weight in the bob and only giving the weight and strength to the rod that are required in order to ensure its rigidity. The whole should be formed so as to enable the pendulum to cut through the air with a minimum of resistance. A lenticular bob, formed of a flat disc thinned off to a cutting edge such that its section through the centre and perpendicular to the surface consists of two segments of circles

facing each other and of such a size that the diameter is to the thickness as 3 is to 2, is found to satisfy these conditions; at least these are the generally accepted dimensions (**1301**).

The resistance of the air diminishes the extent of the oscillations but does not sensibly reduce their period, except when the pendulum is in a very confined case.

As regards the errors occasioned by variations in the density of the air, that is in the barometric pressure, Bessel and Bernoulli regarded them as negligeable, but more recent researches have shown that this is only the case under certain conditions. The amount of the error is about 0·35″ in 24 hours for each inch rise or fall of the barometer, but it depends upon the arc of oscillation of the pendulum; and methods proposed by Airy in England and Redier in France are occasionally used for counteracting its influence.

1285.—While the above advantages are claimed for the lenticular form it has the objection that, if not accurately in the plane of oscillation, the resistance of the air, etc., imparts an undulatory movement to it, the extent of which depends on the density of the air and the greater or less rigidity of the suspension spring at varying temperatures; and this movement may have an effect on the rate in the case of astronomical clocks.

To avoid this possible source of error, especially when employing wooden pendulum rods as they may be twisted on a change of temperature, the lens-shaped bob is often replaced by one that is either spherical or cylindrical.

With a view to determine the resistance of the air, F. Berthoud made experiments on two pendulums of the same weight placed side by side, but one having a spherical and the other a lenticular bob. He demonstrated that the motion of the latter was maintained rather longer than that of the former, but the difference was very slight, about $\frac{1}{14}$-th, and we cannot say whether it was due to the particular form or to the suspension; further experiments conducted with extreme care would be necessary in order to decide this point, which after all is of no importance except in the case of the very finest clocks.

1286.—The following table gives a summary of some of F. Berthoud's experiments. The height of the barometer remained stationary while they were made. A freely suspended seconds pendulum was simply displaced from the vertical and released. The time was noted during which the arc of oscilla-

tion diminished by successive degrees, but we only give the two extreme observations.

Lens-shaped bob weighing 10·4 kilogr.	Arc of	10°	reduced to one of	0·25°	in	29	hrs.	46	min.
	„	0·25°	„	0·12°	„	8	„		
Lens-shaped bob weighing 3·58 kilogr.	„	10°	„	0·25°	„	17	„	6	„
	„	0·25°	„	0·12°	„	4	„		

With the same bob beating half-seconds,
Arc of 10° was reduced to one of 0·25° in 14 hrs. 26 min.

Observation by Berthoud.—"If a half-seconds pendulum perform arcs of double extent so that its velocity and therefore momentum are the same as those of a seconds pendulum with the same bob, it will continue oscillating for nearly as long as the seconds pendulum, so that the resistance due to the suspension is not so great as I formerly believed it to be."

This observation of Berthoud's is important and it proves, perhaps contrary to his intention, that experiments with pendulums of different weights displaced through equal angles can only afford us information as to the resistance of air, because, with a motive force that in practice is usually pre-determined, bobs of different weights will not oscillate through equal arcs nor are their suspension springs identical.

The weight of a pendulum.

1287.—This chapter will be found to be entirely at variance with certain views that were generally accepted as true amongst watchmakers at a time when the science of mechanics was less advanced than it is at the present day, views which have unfortunately been perpetuated in consequence of the neglect of that science by watchmakers.

Thus all the works on horology contain the two following very definite statements, which have been handed down successively from master to pupil:

A pendulum becomes the more capable of counteracting variations in the motive force as we increase its weight;

Long pendulums are preferable to short ones.

The same has been the case in regard to the regulating power of a pendulum, which was always estimated from its momentum instead of its vis viva, a proceeding that would have been justifiable had it been left to itself and not subjected to any external influences; but this could not be the case, and it is necessary to take into account the constant action of gravity as well as the various resistances caused by the air, inertia, shakes,

the action of the escapement, the suspension, the form and bulk of the pendulum, etc.

Two moving bodies that have the same momentum will not give the same force of percussion if they are subjected to different conditions as regards resistance of the air, shakes, the reciprocal action that occurs between moving bodies when near together, etc.

1288.—Consider the case of a clock provided with a light pendulum that makes long oscillations; increase considerably the weight of the bob and observe the effect.

The pendulum first weighs 500 grammes and has an arc of oscillation of 20°, and afterwards it weighs 5,000 grammes and the arc is reduced to 3°.

The weights are as . . . . . 5 : 50
The velocities „ . . . . . 20 : 3.

What will be the resistances opposed to interfering causes by the inertia of these two pendulums?

Calculating by means of the formulæ given in articles **117** and **123**, we shall find that these resistances are, in round numbers:

For the light pendulum 100
„ „ heavy „ . . . . . 20.

The movement of the heavy pendulum will then be interfered with much more easily than that of the other.

With a given motive force, supposed to be just sufficient to drive the clock, the long arc performed by the light pendulum will gradually diminish as we increase its weight until it only just exceeds the lifting arc. As the period of the oscillation has remained the same, the bob will travel in the same time through a much less space and it will therefore move more slowly. As the time occupied by the oscillation is now almost exclusively devoted to the lift, this will take place much more slowly. Hence there will be a double diminution of velocity: (1) of the pendulum; and (2) of the escape-wheel.

1289.—Knowing the weight of a pendulum and the distance traversed in an oscillation, it is easy to ascertain: (1) whether the resistance opposed by the inertia (**123**) has varied; and (2) whether the velocity of movement has increased or diminished; and by repeating this verification for each successive increase in the weight of the pendulum we shall be able to ascertain whether the change of velocity of the wheel is advantageous or otherwise, and we can fix upon the mean that gives, for a given motive force and lifting angle, a maximum effect in

the movement of the pendulum, the least possible resolution of the force exerted by the escape-wheel, and the greatest inertia to overcome resistances.

1290.—These considerations must guide us in modifying the weight of the pendulum and *the motive force should be successively made greater and less;* the erroneous idea that the mere increasing of the weight of a pendulum is sufficient to improve it as a regulator must be set aside (**1298**).

In discussing the several escapements used in regulators and timepieces we have indicated the most usual weight for the pendulum. Aided by these experimental data, any watchmaker will be able to ascertain with ease the weight that secures the best regulator and moderator for a given train.

Impulsive force of the pendulum and force required to maintain it in motion.

1291.—The theory of the pendulum shows that the power causing it to descend when displaced from the vertical, or the impulse that it receives through gravity, is proportional to the sine of the angle of displacement.

The *sine* of an angle or of an arc *a c*, for example (fig. 73, page 700), is the perpendicular *a s* let fall from *a* on the radius *b c*. The portion lying between the foot of this perpendicular and *c*, or *s c*, is the *versine*.

Hence the impulsive force of a pendulum, which is equal to its weight (concentrated at the centre of oscillation) when the pendulum is in an horizontal position, gradually diminishes with the sine of half the angle of oscillation, that is to say the angle measured on either side of the vertical. At 90° the sine and radius become identical.

Let R be the radius and P the weight of the pendulum, A the angular movement on one side, and F the impulsive force (unknown); this force can be determined for any given pendulum by means of the proportion:

$$R : \text{Sin } A :: P : F$$

and we have

$$F = \frac{P \times \text{Sin } A}{R}$$

The radius (*b c*) is known, as it is the virtual length of the pendulum, and the exact value of the sine (*a s*) can be found by trigonometry, or, in practice, by a simple graphical method. Carefully trace out the angle of oscillation on one side, say 40°,

and then draw the circular arc *a c* with a radius equal to the virtual length of the pendulum, say 100 mm. From the point

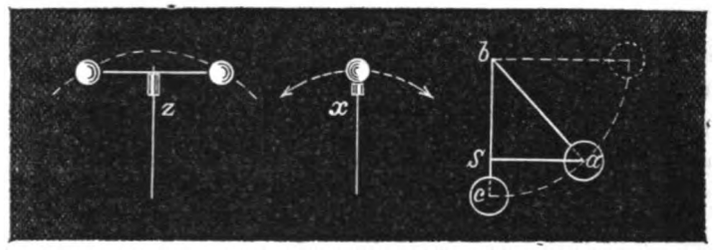

Fig. 73.

a let fall the perpendicular *a s* and measure it accurately. For a radius of 100 mm. *a s* will be 64·28 mm.

Supposing the weight of the pendulum to be 500 grammes we shall have:

$$F = \frac{500 \times 64{\cdot}28}{100} = 321{\cdot}4$$

Thus the force with which the weight tends to descend from *a* is 321 grammes.

If the angle of movement were 90° we should have, since the sine would then equal R,

$$F = \frac{500 \times 100}{100} = 500$$

Or the force exerted by the pendulum is equal to its weight.

On the force requisite to maintain the movement of a pendulum.

1292.—Whatever be the weight of a pendulum, for a given angle of oscillation it will require *the same motive force*, or very nearly the same, to maintain its movement.

It is easy to explain this fact from theoretical considerations: the two pendulums of equal length and with the *same angular movement* can be compared to two heavy bodies falling from equal heights. As gravity exerts the same force on all bodies placed in a given situation, so that *in vacuo* they would fall with equal velocities, it is only necessary to restore the force absorbed in overcoming the resistance of air and of the suspension, for if these resistances could be entirely avoided the pendulum would continue in motion for an indefinite period. This loss is found to be very nearly the same for all pendulums, for if a light pendulum is more impeded by the resistance of air, it has a compensating advantage in the suspension being more flexible.

1293.—If the initial oscillation is produced by the hand, in order to determine the power necessary to maintain the

movement we should require an exact measurement of the difference between the first and second oscillations, or, in other words, the difference between the versines of the two angles of oscillation; from this the amount of force that is lost and requires to be restored could be calculated.

It will be easier to ascertain the falling off in the arc after a certain number of oscillations and divide it by this number, which will give very approximately the loss for each oscillation. We thus know the energy that has to be applied at the centre of oscillation, but the most difficult part of the problem still remains unsolved; for this sustaining power has to be imparted to the pendulum through the agency of a lever and a lifting action, the exact character of which depends on the height and inclination of its plane, the velocity of rotation of the escape-wheel, etc.; and these points are so difficult to calculate that one is always forced to resort to experiment.

Nevertheless, in order that the reader may have a clear idea of the great increase in the resistance as we go farther from the vertical, it will be well to give the following results deduced by Berthoud from his experiments and calculations, although we cannot answer for them from experience.

A seconds pendulum weighing 10·4 kilogr. required, in order to sustain an oscillation of 1°, a weight of 0·122 grm. falling through a distance of 0·75 mm.; and to sustain a movement of 10°, it required a weight of 19 grms. falling through the same distance; thus the 10° required 157 times the force that would sustain an oscillation of 1°.

Pendulums of excessive length.

1294.—Neglecting the various causes, such as the resistance of suspension, etc., that may interfere with or modify its movement, a pendulum, whether long or short, will have the same regulating power if the energy in virtue of its vis viva is invariable. The shorter one will move more rapidly and it will thus compensate for a less weight by an increased velocity.

Theoretically then it is a matter of indifference whether we employ a long or short pendulum.

But observe what practically occurs when we lengthen a pendulum.

In proportion to the total weight, as we continue to indefinitely lengthen it the regulating power of a pendulum gradually diminishes, because, in order to counteract effectively the

trembling of the bob and the flexure and torsion of the rod, it becomes necessary to make the latter so rigid that its mass becomes considerable in proportion to the bob; hence it follows that the centre of oscillation which with a seconds pendulum is very near the centre of figure of the bob will be found to be at some distance up the rod with a very long pendulum, and this will therefore not only fail to satisfy the condition of having a maximum weight with a minimum volume, but it will also present a comparatively greater surface to the air: the effects of changes of temperature will be more detrimental and less easily counteracted, since the entire system will be characterized by excessive pressures and very great tension of the suspension spring.

And, besides these facts, whatever be the length of a pendulum it can only be a good regulator if it entirely masters the motive force and is uninfluenced by changes that may occur in it. For a pendulum of any length will be under the control of the motive force whenever this force is transmitted to it by escapement arms and crutch of sufficient length, that is when applied nearer to the centre of oscillation.

1295.—Having arrived at these conclusions the further solution of the question must be sought for practically. The experience of over a century has shown that the very best clocks, astronomical regulators, which are the most perfect specimens of horological mechanism, do not require a motive force greater than that which will sustain the movement of a seconds pendulum, in other words the force is not so great that the pendulum is incompetent to counteract its irregularities. Experience has further proved that, even with modern turret clocks, it is very rarely necessary to employ a longer pendulum. The seconds pendulum is, then, a limit that had better not be exceeded.

Very short pendulums.

1296.—Commencing from the old rule that a long pendulum secures greater regularity than a short one, we have now demonstrated it to be erroneous so far as regards prolongation beyond certain limits.

Let us now invert the question and, taking the seconds pendulum as a standard, compare it with that beating half-seconds.

In deciding upon the weights and arcs of oscillation it will be observed that the two pendulums can be so adjusted that they offer the same resistance to interfering causes in virtue of

their inertia (**1286**); but in regard to the one beating half-seconds we observe that:

(1) There is a greater angular movement and, in consequence, a greater supplementary arc, which will be proportionately more influenced by variations in the motive force;

(2) The escapement arms being much shorter will require much more care in their construction, etc.

As the dimensions of a half-seconds pendulum are still sufficiently great to admit of such accurate workmanship, it may be relied upon as affording an excellent element of regularity.

Similar reasoning shows that a still shorter pendulum may be satisfactory for ordinary clocks; but in that case the several parts of the escapement have to be very small and it is not an easy matter to ensure that they are well proportioned to the pendulum. It is there that lies the principal difficulty, because we have to see to the freedom of the pivots, the absolute accuracy of the wheel and the length of crutch, which should be very short; the drops will be relatively greater as also the resistance of the suspension.

In a word, a short pendulum moving with sufficient velocity is theoretically as good a regulator as a longer one. If it is found to give inferior results this is due to the fact that the practical conditions that have to be satisfied are more difficult of realization in the case of the shorter pendulum.

On the length of crutch—its suppression.

1297.—In the two last articles when speaking of the length of escapement arm we intend it to be understood that the inter-dependence between it and the length of crutch is taken into account.

We have already referred to this question (**1014**), a question which, to the best of our knowledge, works on horology have never presented in the proper light; we will revert to it.

The object of shortening the escapement arms is to prevent the pendulum being influenced by variations in the motive force: and two cases present themselves.

The impulse arms may be an ordinary anchor movable on pivots, or the pallets may be rigidly attached to the pendulum.

In the first case the escapement arm is in fact a bent lever consisting of an arm of the anchor and the rod of the crutch.

In the second the anchor and crutch are united, forming a simple straight lever.

This diversity of form gives rise in practice to a great number of modes of action on the pendulum.

Since the force of the impulse with a given lifting angle varies very slightly when the length of escapement arm is altered, we will here assume (as we may actually do in practice) that it remains constant. But this force, which we assume invariable with the length of arm, will influence the pendulum very differently according as the crutch engages with it nearer to or farther from the centre of suspension. If the point of application of the force is at *a* (fig. 72, page 694), it will not be resolved in the same manner as if it were at *c*; and the resolution will not occur in the same manner at these points if the weight of bob, or the length or stiffness of the suspension be altered.

We think we have said enough on this subject here and in articles **100-4** and **1014** to prove to the reader that the length of escapement arms and crutch, the length and elasticity of the suspension, and the weight of the bob all form parts of one and the same problem, being intimately co-related. Enough seems to have been said to enable every intelligent watchmaker to solve the problem, and we will conclude with a résumé of the subject.

1298.—From the latter articles and the principles developed in articles **916-934** we conclude as a general rule that: the length of a pendulum depends on the length of its escapement arm (a bent lever when the crutch is used), and conversely; the shortening of the arm that receives the impulse must depend on the motive force, in other words on the pressure which the impulse faces of this arm can sustain without being damaged (**918**); the length of arm that transmits the impulse varies with the weight of bob; and, lastly, the lengths of the pendulum and of the arms that communicate motion to it (the anchor and crutch) should be so inter-related that variations in the force have very slight influence or none at all on the extent and period of the oscillations.

As regards the weight it should be so proportioned to the velocity that the resistance due to the inertia of the pendulum is a maximum; but it is important never to forget that for any given weight of pendulum there is one point at which it should be struck in preference to any other; and that if one changes the other must be altered in proportion, because it is essential to restore the energy of the pendulum in the manner that is least likely to interfere with its free and natural oscillations.

The Pendulum Suspension.

1299.—The earlier pendulums were suspended by a thin cord of some textile substance. Clement, of London, replaced the thread by a long flexible metallic blade, and, by employing two springs instead of one, J. Le Roy rendered the invention of Clement more perfect. If at a suitable distance apart, they prevent the twisting movements of the bob. This is the suspension in use at the present day.

We shall not consider the knife-edge suspension as it is now completely abandoned except for a few rare and very special cases.

If the point at which a pendulum receives its impulse be brought gradually nearer to the centre of rotation, the pendulum being supported by long and flexible blades, at each impulse these blades will have an undulatory movement all the more marked as the point of impulse and the centre of rotation are brought nearer together, the blades made longer and more flexible and the bob lighter. There will result an unintentional shortening of the pendulum when it performs long arcs, and these will therefore be accelerated on the short arcs. This will in part explain the comparative regularity attainable with certain pallet and anchor escapements provided with a flexible suspension of silk or metal and light pendulum oscillating through a long arc; but it is manifest that we cannot regard a blade that is strained beyond reasonable limits as an element of regularity in the going.

Whatever amount of care be exercised in selecting the metal and in making the two suspension springs, we can never be absolutely certain that they are identical; the want of homogeneity in the metal, the hammering, rolling and hardening will prevent it. The watchmaker that copies with the most minute accuracy a suspension employed by a maker of repute will in all probability only be disappointed unless he makes a prolonged series of experiments with it himself, so as to ascertain that the strength and flexibility are suitably proportioned to the rest of the escapement; this will become evident from the rating of the clock.

A metallic blade after it has been worked is generally under molecular constraint, and it must continue in action for some time at different temperatures and supporting the weight of the pendulum in order that it may arrive at its permanent

molecular condition. Some blades in consequence of the bad quality of steel, rough working which has altered the arrangement of the molecules, or bad action of the blades themselves, will never assume this condition. The making of a suspension for a regulator clock will require all the care and attention of which a watchmaker is capable. The adjustment of the two blades in the brass ends by which they are gripped should be perfect, so that they are not strained and hang true; for otherwise the one will act in opposition to the other, and the axis of flexure will be a broken line higher in one blade than in the other. The same will be the case if the springs are of unequal strength, or not hardened and tempered alike, etc., and a badly hung pendulum will be subject to an oscillatory movement that takes it out of the plane of oscillation and is very detrimental. The axis of rotation of the anchor and the axis of flexure of the suspension spring should be in a line.

Some makers replace steel springs by blades of alloyed gold, prepared in the same manner as gold balance-springs.

Isochronism of the oscillations of the pendulum secured by spring suspension.

1300.—In a memoir written by P. Le Roy we read: "This novel observation (concerning the isochronism of springs) may be of great assistance in the adjustment of pendulums, whether small or beating seconds, when the pendulum is supported by a spring; for, in fact, we see that there must be one length of the spring which may make all the oscillations isochronal."

Ferdinand Berthoud, more than twenty years later, appropriating the idea of Le Roy, says: "A well made spring suspension tends to render the oscillations of the pendulum isochronal." Nowhere does he record that experiments have been attempted to verify this statement.

More recently these experiments have been made jointly by MM. Laugier and Winnerl, and they are described in a memoir presented to the Academy of Sciences in 1845.

By employing springs from 1 to 3 mm. long, 5 mm. wide, and 0·24 mm. thick, and adjusting the weight of bob to each length of spring, these observers obtained, with a free pendulum, not merely oscillations ranging from 1 to 5 degrees that were isochronal, but the long arcs were actually more rapid than the short arcs.

Some skilled makers have seemed to fear the great resistance of the springs, which have to be very short, as well as the molecular change that will take place in a spring that is short, thick and, therefore, less elastic.

M. Winnerl replies to these objections by referring to one of his pendulums with isochronal suspension that has gone for eight years without renewal of the oil. The weight of the pendulum was about 7·9 kilogr. (17·4 lbs.) and the driving weight only 1·18 kilogr. (2·6 lbs.).

The authors of the memoir do not consider that there is any advantage in employing the isochronal suspension with the anchor escapement. They confine its use to detached escapements, fearing that the perpetual contact of a tooth against the anchor will constrain the pendulum in its movement, and that the effects of this variable friction, which at the same time modifies both the amplitude and period of the oscillation, cannot be corrected by the isochronal spring suspension whose effect must necessarily be constant.

EXPERIMENTS

ON THE WEIGHT AND LENGTH OF PENDULUMS, THE RESISTANCE OF THE AIR, AND THE MOTIVE FORCE ABSORBED IN THEIR MOVEMENT.

(From a work published by M. Wagner.)

1301.—The articles relating to the pendulum were in the press when a small work issued by M. Wagner at the Exhibition of 1867 came under our notice; in it he describes the series of experiments he has made on detached pendulums and the conclusions that he has drawn from them.

We shall briefly summarize the principal portion of this work without discussing it or expressing any opinion, and for the author's explanations must refer the reader to the original work.

"The resistance opposed by the air to the movement of a pendulum is generally proportional to its surface and to the space through which it moves. With pendulums of the same length and weight, the resistance of the air is proportional to the square roots of their surface. The sphere (as compared with the lenticular form, etc.) is the best possible shape, since it contains the greatest amount of matter within a given enveloping surface." The author then concludes that the air offers a minimum resistance to round bobs (**1285**).

"With pendulums of the same length and surface and equal

amplitude of oscillation the amount of the motive force absorbed in each oscillation remains precisely the same whether the pendulums weigh 1, 2, 3, 4, 8, or 10 kilogr." (**1292**).

"The going of a clock is rendered more regular by increasing the weight of the pendulum, and the force required to maintain the movement is independent of this weight; hence it is evident that heavy pendulums should be preferred to light ones." The author makes no reservation as regards excess of weight except that "it must be proportioned to the spring, knife-edge, or pivots that have to support the pendulum." It is important to keep in mind the fact that his pendulums always oscillate through the same arc (**1288** and end of **1286**).

"The suspension offers a resistance to the movement that increases with the weight of the pendulum." The author considers that this increase in the resistance is proportional to the weight, in other words to the tension of the suspension spring.

"My experiments show that the length of a pendulum should be something between 0·25 metre and 1·50 metres" (**1295**).

"These experiments lead to this remarkable conclusion; that, whatever be the amplitude of oscillation, the force absorbed by the friction of the escapement during each oscillation is always strictly proportional to the energy possessed by the pendulum in virtue of its movement."

"The force absorbed by friction is in proportion to the arc traversed."

We pass over the portion of the memoir that treats of the weight of annular balances, their losing rate in extreme temperatures, etc. It is obvious that the author, who is very skilful in his speciality, turret clocks, has never practically studied the chronometer and is unacquainted with the work of horologists of the present century.

CONSTRUCTION OF COMPENSATION PENDULUMS.

1302.—Compensation pendulums with 9, 5, and 3 rods and various other arrangements more or less resembling that discussed in paragraph **1270** will be found described in most horological works, for example Reid's *Clock and Watchmaking*. To this we refer the reader, only observing that by means of tables of dilatation, etc., it is easy to calculate the approximate dimensions so as to plan the general disposition of any given

pendulum; it is especially necessary that the surfaces subjected to pressure be sufficiently large without needlessly increasing the thickness of the rods, which, if thick, take too long to be affected by changes of temperature, and if thin are acted on promptly but bend or distort at the surfaces of contact.

It is important to employ metals that are thoroughly homogeneous and whose force of contraction and dilatation have been accurately determined (**1261**).

It also would be well to know whether the regulator, when permanently located, will be subjected to appreciable differences of temperature throughout its height, due to the mode of warming or exposure to the sun, as in that case it would not be a matter of indifference whether the gridiron or mercury pendulum is adopted.

The details given in the articles on compensation, etc. (page 682), will render further explanation unnecessary.

1303.—*Mercury Pendulum.*—The earlier mercury pendulums never had more than one vessel to contain this metal, but they are now sometimes made with the mercury distributed between two or more parallel tubes. This arrangement, which is due to Duchemin, involves more labour and care in order to ensure a perfect adjustment, but it has the advantage of displacing the centre of gravity of the liquid less, and is also more rapidly affected by changes of temperature.

To calculate the proportions of a mercury pendulum.—If we assume the centre of oscillation to be coincident with the centre of gravity of the bob (which is very near the truth, since the mass of the bob is very great as compared with that of the rod), taking L as the length of the rod, and K the coefficient of linear dilatation of the substance of which it is made, L K will be the expansion for 1° C., in other words the space through which the centre of gravity of the mercury is lowered.

If the mercury occupy a height h of the tube, its dilatation will at the same time cause this centre of gravity to ascend through a distance equal to $K' \frac{h}{2}$ (K' being the apparent coefficient of linear dilatation of the column of mercury deduced from the apparent cubical dilatation, $\frac{1}{6480}$, as given by Dulong and Petit). We thus have the equation $L K = \frac{h}{2} K'$ for determining the height H of mercury column that will secure perfect compensation.

Assume $\pi r^2 h$ to be the volume of the cylinder of mercury

($\pi = 3{\cdot}1416$); its cubical dilatation for 1° C. will be $\pi r^2 h \frac{1}{6480} = K'$, and the above formula thus becomes:

$$L K = \frac{h \pi r^2 h}{2\ 6480} = \frac{\pi r^2 h^2}{2 \times 6480} = \frac{r^2 h^2}{4125}$$

The vessel may be made of either glass or iron, but glass is fragile and a bad conductor. Iron being a good conductor makes the action of the compensator more prompt.

Notes completing this article will be found at the end of the volume (**1482–3**) as well as the description of a mercury pendulum designed by M. Vissière.

1304.—*Observation.*—In consequence of capillarity (**89**) the free surface of the mercurial column assumes a form that is more or less convex, and some horologists have feared that this would be a source of error in the compensation. If the form always becomes precisely the same at any given temperature the variations of the degree of convexity will not affect the compensation; but it does not follow that such will be the case if, through impurity in the mercury or the character of the sides of the vessel, the effect of capillarity is variable; in other words if the mercury does not always return to the same level at any given temperature. We are not aware that anyone has ever, in experimenting on pendulums made by the best makers, succeeded in detecting appreciable variations due to different degrees of convexity on a given change of temperature.

On the slide adapted to the pendulum by Huyghens.

1305.—The theory of the slide, a small mass of metal that can be fixed at varying positions on the pendulum rod, was enunciated by Huyghens; but it is not suitable for insertion here and we would only observe that by moving the slide upwards or downwards the distance between the centres of suspension and oscillation can be varied when regulating the clock, in cases where the error to be corrected is so minute that there would be danger of going too far if we attempted its correction by moving the screw that supports the bob.

The advantage of the slide consists in the fact that we can modify the position of the centre of oscillation by an almost inappreciable amount by moving it through comparatively large distances.

Take the pendulum *s o* (fig. 72, page 694) and on fitting a slide to it at *t* its oscillations will be more rapid; on raising the

small mass to h they will be still quicker. If it be at g, a point near the middle of the rod which has to be determined, the accelerating effect will be a maximum. If it be raised through a space $g\ i$ equal to $g\ h$, the rate of the clock will be the same as if it were at h. Hence, whether it be made to ascend or descend from the point g, there will be the same retarding effect.

This action is not proportional to the interval through which it is moved. For two equal ascending movements commencing at the points t and h, although the acceleration be marked at t it will be hardly sensible at h. The effect of a given displacement is all the greater according as the slide is farther from g, so that it is well not to bring the extreme point of its path too near to this point g.

The slide is a very useful appendage for completing the timing of astronomical clocks, but it can only be of very limited use in ordinary clocks where the variation that occurs in twenty-four hours is often much more than it can be expected to correct. Its effect is completely swallowed up in such a variation.

Some watchmakers have applied it to ordinary clocks, and it failed to give the results they anticipated; this is in no way surprising. What is astonishing is the extraordinary conclusion that some of them have come to from their want of success, when they could actually write that "without doubt Huyghens made a mistake"!

The pendulum prolonged above its centre of suspension.

1306.—If we prolong a pendulum above the centre about which it rotates the oscillations are retarded, an effect that is perfectly explicable on theoretical considerations and is all the more marked as this prolongation is made heavier. It thus becomes possible to diminish the number of oscillations in a given time by extending the pendulum above the centre of rotation and fitting a sliding weight to the upper portion.

M. de Prony proposed to complete the adjustment of the pendulum by employing two balls attached to a horizontal bar that turns with friction on the upper end of the pendulum (fig. 73, page 700).

It will be evident from the figure (at z and x) that the two weights will act together and with equal force, as though there were only one ball of twice the weight on the pendulum rod, when they are set in the plane perpendicular to the plane of

oscillation (x), whereas if the bar is set at right angles to this position or in the plane of oscillation as at z, its action as a whole will vary in intensity according to the positions occupied by the balls, and the two will tend to modify the period in a similar manner; the effect on the pendulum will be very different in these two extreme positions and will gradually increase as the bar is moved from the first to the second.

The prolongation of a pendulum above its point of support enables us to cause it to beat a less number of oscillations in a given time, so that they may be as slow as required; but it is then necessary to make the sliding weight very heavy or else the prolongation considerable, either of which will introduce inconveniences in the suspension and compensation and in the details of construction; it is moreover nothing but a step towards converting the pendulum into an annular balance, and it will gradually come to resemble this latter in being sensitive to variations in the motive force.

This device cannot be employed in high-class horology (at least not as a reliable regulator), but Maelzel made a very interesting application of it in the metronome, an instrument for beating the time of music.

THE ANNULAR BALANCE.

Historical notice.

1307.—The first balance employed was termed a *folliot* (shown on page 55) and carried two small suspended weights or *régules* which could be moved to or from its centre according as it was required to accelerate or retard the movement of the clock. The folliot employed in watches consisted of a metallic rod terminating at either end in a heavy mass, the whole being formed out of a single piece of metal. In some old clocks the angular path of the folliot was limited by two pins, the position of which could be varied, and it was by bringing them more or less together that the acceleration or retardation was effected; but in the great majority of the old watches we have examined there is a small annular balance acting precisely like that of a verge watch when the balance-spring has been removed.

Since a balance contracts in the cold and expands in heat its oscillations will become more rapid on a fall of temperature

and *vice versâ*. In order to counteract this source of error as well as others due to the balance-spring, etc., Harrison suggested the use of a bi-metallic strip that altered the acting length of this spring when the temperature varied.

The method was condemned by P. Le Roy as involving a denial of the principle of isochronism, and it was then that this ingenious man laid down the principles of the compensating balance employed at the present day. On examining his work and the figures that illustrate it we find three arrangements:

(1) An ordinary annular balance with two heavy masses moving to or from the centre on a change of temperature through the action of bi-metallic arcs attached to the plain rim of the balance.

(2) A balance with bi-metallic divided rim similar to those in use at the present day but having four arms and weights.

(3) Lastly, a balance with a plain uncut rim of a single metal, but carrying two small thermometers on opposite sides, which effected the compensation by the dilatation of the mercury. This balance was moreover provided with two timing screws as in the modern chronometer.

P. Le Roy when he made his two chronometers adopted the last arrangement, which is very easy of construction, and gives, without any contacts or pressures on the acting parts, a rectilineal movement of the compensating masses; whereas the two others, especially when we consider the large dimensions of the balances, presented serious difficulties of construction.*

Le Roy's chronometer was exhibited to the king and to the Academy of Sciences in 1766, and the memoir describing it was also transmitted to the Academy in the same year. It was published in 1770.

Two years later J. Arnold made chronometers with compensation balances in accordance with Le Roy's principles, having a cut bi-metallic rim with compensating weights and timing screws.

The earlier compensation balances were very difficult of construction as they were formed of two metallic blades which

* P. Le Roy gave the impulse to the balance at its circumference, a plan that has been unfavourably criticized. Before the impulse near the centre could be thought of it was essential that experience should demonstrate the fact that the balances of chronometers could be reduced to the dimensions then adopted for large watches, and this was contrary to the opinion of men of science and watchmakers of that day. They insisted on the superiority of large balances. The mistakes committed by P. Le Roy, which are rare, are due to the condition of science at his time and more especially to the want of experimental data. Chronometry was then in its infancy.

were adjusted to fit each other and then fastened together by innumerable minute rivets. Many that are to be seen in chronometers by L. Berthoud and A. Breguet are perfect models of workmanship, and they are not surpassed in beauty and perfectness by any mechanism of more recent date. At the present day one metal is melted on to the other.

THEORETICAL CONSIDERATIONS AND DEFINITIONS.

Moment of inertia.—Regulating power of balances.

1308.—The *moment of inertia* of a balance or of any body whatever rotating on an axis is the sum of the products of each infinitesimal portion into the square of the distance of that portion from the axis **(49)**.

It amounts to the same thing as if we consider each particle to be performing an independent movement at the extremity of its own radius; but, since all the molecules are rigidly connected together, the greatest force will be exerted at the extremity of a radius of such a length that the action of the molecules that lie beyond its extremity is equal to that of those that fall within it. At this point the action will be the same as though the entire mass were concentrated at it.

Through this point passes the *circumference of gyration* of the balance, and the line joining it with the centre of rotation is termed the *radius of gyration;* in other words it measures the virtual size of the balance. The diameter of gyration of an annular balance is analogous to the simple pendulum corresponding to a given compound pendulum; it is in fact a *mean* diameter.

1309.—The moment of inertia is a measure of the resistance that the balance can oppose to causes that would modify its movement, and, with a given mass, its value increases or diminishes according as we move the mass from or to the centre of rotation.

The *regulating power* of a balance then is not measured by its external dimensions or total weight, but depends mainly (according to the most generally accepted opinion):

(1) On the amount of its moment of inertia.*

* It would then be an error to take the momentum of the balance as a measure of its regulating power, for to disturb or arrest its movement it is essential that we partially or entirely neutralize its *vis viva* and this is represented by the product of mass into the square of the velocity.

(2) On the manner in which its form renders it insensible to external influences, friction, &c.

1310.—*With a given mass or weight, the moment of inertia is proportional to the square of the diameter or radius of gyration, and these radii are inversely proportional to the number of oscillations in a given time;* that is to say the accelerating or retarding effect produced by bringing the mass towards or from the centre of rotation varies with the square of the distance between this centre and the centre of the mass, or, in other words, varies inversely with the number of vibrations.

1311.—*With a given radius of gyration the moment of inertia is proportional to the mass or weight.*

The squares of the numbers of vibrations in a given time vary inversely with the moments of inertia.

Hence it follows that *with a given radius the masses are inversely proportional to the squares of the numbers of vibrations;* or, in other words, *inversely as the squares of the times indicated by the hands.*

1312.—These considerations prove that we can have no exact knowledge in relation to the regulating power of a balance so long as we only regard its external dimensions. This is why the discussions as to the relative advantages of large and small balances in watches, that are so frequent among watchmakers, never have and never can result in any real good.

1313.—*Observation.*—The word "velocity" which is so frequently employed may give rise to confusion unless the two kinds are clearly distinguished.

(1)—*Angular velocity* is the angle through which an arm turning on an axis is displaced in a unit of time. It is entirely independent of the length of this arm. The approximate ratio of the angular velocities of the balances with the cylinder and (pocket) chronometer escapements in the same unit of time

We have elsewhere (**661**) observed that in practice we approximately estimate the power of a balance at the moment of unlocking the escapement by its momentum. This mode of obtaining an approximate value is not opposed to what precedes, as will be seen when it is remembered that, at the instant under consideration, the balance and balance-spring are moving together with a previously acquired velocity, and it results from the experiments already described that a considerable amount of energy is wasted through the impact and elastic reaction. The unlocking only occupies a minute portion of the entire movement of the balance, which is, during almost the entire vibration, either under the influence of the balance-spring or being accelerated in consequence of the lifting action.

A complete solution of this question could only be arrived at by the employment of mathematics of an advanced description and by making further experiments (**1383**).

(one-fifth second when there are 18,000 vibrations per hour) is about 270° : 360°.

(2)—The *velocity* properly so called is the space traversed in a unit of time by the point under consideration (which in this case is taken on the circumference of gyration). For a given angular movement we obtain the approximate ratio of the velocities by multiplying each radius by the number of vibrations in a unit of time.

To determine the radius of gyration and moment of inertia of a balance.

1314.—Since the *size* of a balance, that is the only size that can be taken as a basis of calculation, is represented by twice the radius of gyration, and since we know that the regulating power depends on the moment of inertia, and that, with a given mass, the moment of inertia varies as the square of the radius of gyration, it follows that we must of necessity commence by ascertaining this radius of gyration of the balance. The exact calculation is very difficult; but a sufficient approximation can be obtained by the following methods for all ordinary purposes.

In the chronometer balance, which is loaded with heavy masses at the circumference, the centre of gyration is a trifle nearer the centre of figure than the centres of these round weights; and, to approximately ascertain the moment of inertia, we take the distance between the axis and the middle point of the rim (or, with prismatic compensation weights that are very wide at their outer edge, measure from the external surface of the rim) as the radius of gyration, and obtain the moment of inertia from the following formula:

$$A = \frac{P \times r^2}{g}.$$

A is the moment of inertia, P the weight of balance, r the radius of gyration, and g the value of gravity (**48** and **126**) which may be replaced by the number 9.80896 (on the metric system, or 32.2 in English measure).

That is to say, multiply the radius by itself and the product by the weight of balance. This product is divided by 9.80896; and the result so obtained is the moment of inertia of the balance.

Employing this number to represent the value of gravity, the weight P should be given in *grammes* and the radius r in *metres*.

1315.—If we proceed in a similar manner in the case of watch balances, where the metal is very unequally distributed in the rim, the results will be too far from the truth to serve any useful purpose.

The following method is applicable to all cases. Take a balance exactly like the one under consideration as regards weight and dimensions, and separate the arms from the rim by cutting through their extremities. Then weigh separately and with great care the rim and the three arms, which are still united at their centre, and the radius of gyration will be given by the formula:

$$K = \sqrt{\frac{p'}{P} \times \frac{R'^2}{3} + \frac{p}{P} \times \frac{R^2 + R'^2}{2}}$$

in which $p + p' = P$. We neglect the weight of the small disc at the centre of the balance. K is the radius of gyration,—P the total weight minus that of the centre disc,—p the weight of rim,—p' that of the three arms taken together,—R the external radius of the rim,—R′ its internal radius.

Practical Applications.

1316.—*A balance beats a certain number of vibrations in a given time; how much must its weight be increased or diminished in order that it may lose or gain a given amount in that period?*

First Example.—Consider a balance that loses 5 minutes in an hour and weighs three grammes; we get the proportion: The square of the time indicated (55 minutes in an hour) is to the square of the time required (60 minutes), as the unknown weight (x) is to the known weight (3 grammes); or $55^2 : 60^2 : : x : 3$,

$$\text{that is } 3025 : 3600 : : x : 3,$$

$$\text{whence } x = \frac{3025 \times 3}{3600} = 2{\cdot}52 \text{ grammes.}$$

Thus the weight of the balance must be diminished by 48 centigrammes.

Second Example.—Now assume the balance to still weigh 3 grammes but to cause a gain of 5 minutes, so that it indicates 65 minutes in an hour.

The proportion will be

$$65^2 : 60^2 : : x : 3$$

whence we obtain $x = 3{\cdot}52$ grammes.

So that the weight must be increased by 52 centigrammes (the milligrammes are neglected).

1317.—*If a balance* (B) *make a known number of vibrations, find how much it will be necessary to increase its diameter* (*of gyration*), *the weight remaining the same, in order that the new balance* (*b*) *may make any other given number of vibrations.*

This problem will be solved by the formula (**1310**):

Radius B : radius x : : number of vibr. of b : vibr. of B.

Example.—Let B have a radius of 10 mm. and make 15,000 vibrations in an hour; what should be the radius of b in order that it may make 15,500 in the same time?

$$10 : x : : 15{,}500 : 15{,}000.$$

whence

$$x = \frac{10 \times 15{,}000}{15{,}500} = 9.67.$$

Thus the new balance b must have a radius of gyration equal to 9.67 if it be required to perform 500 vibrations more than the other in the given time; or it must be reduced by one-third of a millimetre.

If the weight is the same when the two radii are given as well as the number of vibrations of one, the same proportion would be used to determine the other number of vibrations, except that x would be replaced by a known radius—and the unknown quantity would be the number of vibrations of, say, b.

1318.—*Observation.*—In these several cases we have only considered the changes in the number of vibrations that correspond to changes in the weight and diameter, and have not referred to the increase or decrease in the regulating power. But this is a separate question which must be solved by comparing the moments of inertia (**1309**).

We would again observe in conclusion, since experience has shown that we cannot repeat too often when it is our object to put a stop to deep-rooted prejudices and objectionable modes of procedure, that it is impossible to reason legitimately on the subject of balances unless the radius of gyration is taken as a basis of calculation.

METHODS THAT HAVE BEEN PROPOSED FOR ASCERTAINING THE SIZE AND WEIGHT OF A BALANCE.

Size of balance.

1319.—Moinet's Treatise says very little on this subject and what there is is little more than a paraphrase of what Jurgensen had already copied from Berthoud: "The weight and diameter of the balance must be proportioned to the motive force and the number of vibrations." Having said this they proceed to give a number of callipers in which balances of different dimensions give the same number of vibrations in a given time, and the two authors arrive at the same conclusion as Berthoud but in a somewhat different form: "There is a happy mean which experience has proved to be correct, and by copying the dimensions of balances by the best makers that have been shown to have a good rate, we shall have an easy and certain guide."

1320.—An objection to these instructions certainly was raised by Perron: "If," he said, "the size of the balance is proportioned to the number of vibrations in a given time, we should require a balance of the same size in watches whether they are large or small," and this he declared impracticable. This objection of Perron is not of much importance, and yet it was tacitly accepted because no one contradicted him.

1321.—It has been proposed to take, as a measure of the balance, five times the height of the mainspring.

Two Paris watchmakers have pretended that if a pendulum be coiled into a circle it will give a balance beating the same number of vibrations. The diameter of the balance will then be rather less than a third the length of the pendulum; but this so-called principle will not bear examination any better than the one that precedes.

1322.—An ingenious watchmaker, M. A. Vallet of Bordeaux, has observed that in a number of chronometers the most satisfactory rates were given by those whose balances had a diameter, measured across the rim, equal to about two-thirds of the sum obtained by adding together the radii of the three wheels and of the escape-wheel. This is a useful experimental datum that is applicable to the calliper ordinarily met with.

1323.—Lastly it is the general practice to accept it as a rule (which however certain exceptions do not tend to confirm)

that the *size of the balance* is given by that of the *circle covered by the motor*, which is determined, according to some, by the diameter of barrel and, according to others, by that of the spring when coiled up; this practice was adopted by A. Breguet and the English makers who have adopted a chronometer by Earnshaw as a model. The diameter of the barrel cover is a mean between the above dimensions and is therefore generally dignified by being taken as the measure.

1324.—Some horologists who do not accept this as a basis of measurement have pointed out that the force which it is important to know is that exerted by the escape-wheel; with them, then, the *motor circle* is this wheel, and they make the diameter of the balance from two to two and a-half times its diameter.

Observation.—In some of the plans that have been proposed useful methods of approximation are to be met with, but it is impossible to found a theoretical law and a universally applicable rule on any of them. We thus see that the usual practice amounts to nothing more than an imitating of balances that have given satisfactory results, and a theoretical starting point is absolutely wanting.

To find the weight of a balance.

1325.—As regards the determination of the weight of balances the ordinary practice amounts to little more than this: copy one that has given satisfactory results, increasing or diminishing the weight in accordance with the greater or less facility of starting and the differences observable in various positions. The motive force in this case is assumed to be previously fixed upon, as usually happens with ordinary watches; but in the case of chronometers the question of the weight becomes complicated with that of isochronism, and it is essential to *feel* the best proportion, causing the weight and motive force to vary, according as the rate and the ease with which the lift commences indicate to be necessary.

ELEMENTS OF THE THEORY OF THE ANNULAR BALANCE.

Preliminary Considerations.

1326.—It is evident from what precedes that the methods ordinarily adopted for determining the proportions of an annular balance are founded on no theoretical basis; the whole practice

amounts to the examination of such as have been proved satisfactory and then copying them, increasing or decreasing the dimensions according to the kind of escapement that is being made.

The subject we proceed to discuss is then quite new. If the solution that we have to offer is not as complete as could have been wished, we shall at any rate have put on the track of a strict mathematical solution of the problem those of our fellow-horologists and men of science who have more leisure than we have; and it is to be hoped that some of them will consider the satisfaction of rendering a service to horology a sufficient encouragement to undertake the work. This is the only recompense that we have secured after fifteen years of difficult work, the results of which were presented at the Exhibition of 1867.

1327.—We will begin with a proposition that does not seem to need demonstration.

All the movements that are met with in nature are governed by laws which collectively constitute the science of mechanics in its widest sense. It proves to us that these movements cannot take place with increased velocity or pressure without introducing disturbing elements; but unfortunately many of these interfering causes escape our means of observation since they have their source in: elastic actions and reactions; a vibratory movement that is more or less intermittent; permanent or temporary changes in the molecular state; production of heat; electric action; effects of centrifugal force; excessive motive force occasioning excessive pressures, etc., etc.

If we cannot always appreciate these causes we can at any rate obtain evidence of their effects: thus if a pendulum be subjected to such a force that it makes a different number of oscillations from that which the tables indicate for its virtual length, the regularity of the movement will be disturbed, in other words the pendulum will vary in its rate either constantly or at irregular intervals.

The same is the case with an annular balance when it varies much from certain recognized dimensions. Every watchmaker must know that it is, in many cases, only necessary to replace the balance of a watch by one that is rather larger or smaller to entirely avoid, or at all events diminish, irregularities observed in the rate which could not be got rid of by any other means. It is said that when watches were formerly made to beat more

than 20,000 vibrations in an hour, timing was impossible. The same is the case at the present day if the dimensions differ materially from what is generally recognized as the best.

1328.—We must leave pure science, which, so far as we are aware, has not considered this subject, to seek for an explanation of these facts; it is enough for us to know that they are so well established as to afford practical evidence that the maximum of regularity in the oscillating movement of a mass depends on a ratio that has to be determined between the length of the *radius of gyration* and the number of oscillations in a given time.

Starting with a balance that has been proved to be satisfactory with a given number of vibrations, we can calculate the most suitable dimensions of any other balance with any other number of vibrations; and, since the same laws govern the movements of a pendulum and balance, we can with certainty deduce this curious result, that if we ascertain what length of pendulum best satisfies the conditions that govern the annular balance, it will amount to determining the law of increase or decrease in the balance, and the table of the lengths of simple pendulums will constitute a scale by which it can be measured.

It seems reasonable to hope that the question thus stated might be solved by the higher methods of analysis if undertaken by an accomplished mathematician. In the absence of such a solution, and since exact figures are not absolutely essential in practice, we will content ourselves with adopting the experimental method to which it is always necessary ultimately to have recourse.

EXPERIMENTS

That have served to ascertain the relation between the pendulum and the annular balance.

1329.—If an ordinary clock that is controlled by a pendulum be carried about, the conditions of the movement of its regulator will be so far modified that its rating will be entirely destroyed. It would be easy to explain the physical and mathematical reasons for this, but such an explanation would be utterly superfluous as the fact is so well known.

A study of the causes of this irregularity led us to inquire whether it was not possible to construct a clock controlled by a pendulum that would be uninfluenced by a movement of translation; and a reply to this inquiry was furnished at once by the two following laws:

(1) A body without inertia *would be indifferent to a state of rest or motion.*

(2) A moving body resists a change in the direction of its motion all the more *according as its movement is more rapid.*

It was then only necessary to experiment upon a moving pendulum, gradually diminishing on the one hand the resistances that opposed an acceleration in its movement (by reducing its mass and radius of rotation), and proportionally increasing, on the other hand, its velocity or the number of oscillations in a given time.

In the experiments we made, and which we should have repeated had we had time, we were never able to establish, with any degree of certainty, an indifference to the movements of translation until the pendulums were made very short, more especially with those beating 30,000 vibrations per hour and upwards.

This result furnishes us with a measure of the inertia and the velocity that a mass, oscillating at the extremity of an arm, should possess in order that it may not be sensibly affected by movements of translation. Having arrived at this point, the simple pendulum becomes a standard from which we can deduce all the diameters of gyration of the annular balances employed in watches and chronometers.

The pendulum that determines the size of a balance.

1330.—Let us consider the simple pendulum that gives 29,343 oscillations in an hour, remembering that this figure is only exact when the oscillations are very small. As soon as the arc becomes more extended, the oscillations will occupy a longer period. Hence there would be gradually increasing loss on the rate as the amplitude of the arc becomes greater. Assume the arc to measure 270°.

From a formula given by Poisson in his *Traité de Mécanique* we calculate that in the case under consideration the loss on the rate would be 44,000 seconds in 24 hours or 86,400 seconds; in other words, a loss that will make the actual number of vibrations in an hour, compared with the number given in the table, as 424 : 864. Hence this short pendulum measuring 14·8 millimetres would in reality only beat 14,400 vibrations in an hour.

1331.—This small pendulum will be very insensible to variations in the motive force on condition that it is retained in

a vertical position. If we desire to make it capable of also withstanding any changes of position whatever, its rod must be prolonged as far above the point of suspension as it extends below, and the pendulum bob must be divided equally between the two extremities. If, when in this condition, an elastic force act at its axis in such a manner as to bring it back to the position from which the motive force has impelled it, just as gravity itself does, we shall have secured the balance which, for a certain number of vibrations, affords us most certainty in the timing, since it possesses *in a higher degree than any other* the inertia and velocity that are known to be best capable of counteracting the irregularities in the rate caused by movements of translation; and further, we have in the balance-spring a power that can, within certain limits, be modified as required.

Theoretical sizes of balances.

1332.—From these experiments we conclude that: For a given amplitude and period of oscillation, the radius of gyration of an annular balance is the same as the length of the simple pendulum that beats the same number and oscillates through an equal arc.

The following table is calculated on this basis, taking a mean arc of 270°.

PENDULUM.		BALANCE.	
LENGTH. mm.	OSCILLATIONS PER HOUR.	DIAMETER. mm.	VIBRATIONS PER SECOND.
239.0	3,600	478.0	1
60.0	7,200	120.0	2
26.5	10,800	53.0	3
14.8	14,400	29.6	4
9.6	18,000	19.2	5
6.6	21,600	13.2	6

The proportions observed in marine and pocket chronometers that have secured good rates, which will be subsequently given, will afford evidence that this is the true rule, the theoretical starting point for determining the diameters of annular balances; and this is the sole element that was wanting, for the weight is known since it depends on the motive force that impels the escapement.

1333.—It is important to remark that this table gives the diameter of gyration of a balance that is not subjected to any resistance tending to check its movement; the diameter is

a maximum. Just as the length of a simple pendulum must be reduced when it is replaced by a material pendulum mounted on pivots, owing to its being under a retarding influence, it will be evident that, in practice, the theoretical diameter of a balance must be reduced to correspond with the retarding forces, which are greater with one escapement than another.

These causes are:

(1) Friction of the pivots.

(2) Lifting action more or less disturbing the movement.

(3) Friction on resting faces that are concentric with the axis.

(4) Lateral pressure of the pivots due to the excentric action of the balance-spring.

(5) Resistance of the air and of unlocking.

1334.—For a definite number of vibrations (say 18,000 for example):

The chronometer escapement has only a lifting action at every two vibrations, and its pivots are usually made very fine, while its balance-spring developes concentrically; hence the diameter of gyration of its balance is very little less than the theoretical diameter.

As compared with the preceding, the lever escapement has heavier pivots, one lift at each vibration, etc., so that the causes of loss are more marked and the actual diameter should be still further reduced.

Lastly the cylinder escapement, regarding the mass of its balance, has very large pivots, a considerable and continuous friction on the axis, and a short balance-spring which materially increases the lateral pressure of the pivots; the movement of the balance will then be much impeded and in order that it may have the requisite liveliness, the radius of gyration must be reduced from what theory indicates in proportion to the reduction in the number of vibrations, a diminution that is caused by these resistances.

1335.—It has been seen that the regularity of movement of the balance will be modified in a certain definite proportion to correspond with these resistances and will be especially influenced by their variability. It is for this reason that the cylinder escapement cannot be timed as accurately as a chronometer, and the chances of uniformity in the rate will become greater as we diminish the resistances and thus approximate more and more to the theoretical radius.

EXPERIMENTAL DATA.

1336.—In the case of the greater number of pocket chronometers beating 18,000 vibrations that were characterized by excellent rates, so far as we have been able to determine their dimensions, the radius of gyration was about 9·3 millimetres. The deviation from the above table (0·3 mm.) gives an approximate measure of the retarding influences. This difference would have been greater because the arc of vibration exceeded 270°, but the excess was counteracted by the isochronism of the balance-spring.

The balances in some lever watches of J. Jurgensen with 18,000 vibrations, the uniformity of whose rate has not been surpassed, had a radius of gyration of about 8·7 mm. The difference due to the retarding influences was then about 1 mm. or rather less.

Lastly there are excellent cylinder watches in which the radius of cylinder is to radius of gyration of the balance : : 1 : 15, the radius of gyration being about 7 mm. The difference due to retarding influences is about 2·5 mm.

By repeating several times over our first experiments, taking every possible care, it would be possible, after having corrected and repeated them, to deduce the elements of a formula for calculating the most suitable mean balance for any given escapement; but it would not be much use unless we possessed sufficiently exact information as to the amount of the friction in escapements, of which we are at present ignorant. The figures given by Berthoud and Romilly* are often at variance not only with the laws of friction as given in treatises on physics but also with experience.

1337.—Abstruse researches in horology are so little encouraged at the present day, when the re-inventors of old-fashioned toys are almost the only ones that are permitted to attract attention, that it is very doubtful whether the delicate and difficult experiments which this subject involves will be again taken in hand. Only one horologist of our acquaintance has had the courage to undertake them, and he had the knowledge necessary to bring them to a successful issue; but, wearied

* This horologist was born at Geneva in 1714, and died in 1796. He published some interesting articles in the *Encyclopédie*, and acquired a high reputation in Paris, where he passed the greater part of his life. He advocated watches that went for eight days without winding, but was not very successful with them.

by the indifference of others, and the absence of all assistance from watchmakers that possessed fortune or reputation, as well as from men of science, he gave them up; and this is greatly to be regretted for our art.

Owing to the pressure of other business we have ourselves also been compelled to give up the study of the subject, but as others, more fortunate than we have been, may be induced to complete what we have commenced, we would request that after having repeated the experiments already referred to with the utmost possible care, they will make the following observations.

Experiments that remain to be made.

1338.—Carefully construct a chronometer balance with a plain metallic rim and the theoretical radius of gyration, and fit to it two staffs that are absolutely identical except that the pivots of one are very conical and carefully hardened at their extremities. Fix the balance to the first staff, attaching to it a balance-spring with a theoretical terminal curve taken from a chronometer whose balance is of the same weight and dimensions as the one under experiment, making, say, 14,400 vibrations per hour. Then mount the experimental balance vertically between two very hard jewels, in which very shallow holes are marked. If well poised the balance will oscillate for a long time, as the resistance due to friction is reduced to a minimum.

Count several times over the number of vibrations in a given time.

Substitute the second staff for the first and mount the balance precisely as it would be in a chronometer.

Count the number of vibrations, first in a vertical and then in a horizontal position.

The loss will give a measure of the friction and will indicate the amount by which the radius of gyration should be reduced on account of the friction of the pivots.

1339.—These preliminary experiments should be supplemented by the following:

At the surface of the staff (whose diameter bears a known ratio to the diameter of gyration) apply a friction analogous to that of the escape-wheel, and, after having determined the loss so produced, repeat the experiment, using two or more discs of gradually increasing diameters adjusted on the axes, observing the successive retardations. Such determinations will afford a

first approximation in fixing the radii of gyration of the balances of frictional rest escapements.

Different impelling forces should also be applied.

If in these various experiments the movement of the balance could be maintained by the agency of a very uniform motive force, and at the same time indicate hours, minutes and seconds, a greater degree of accuracy could be attained, because they could occupy a longer period; then the temperature should be maintained invariable during their performance.

We shall not consider this subject further; if fully treated it would lead the way to the acquisition of much new information in the horological art.

WEIGHT OF THE BALANCE.

Its relation to motive force.—Its distribution throughout the different parts.

1340.—With a given radius of gyration, the weight should increase with the impelling force, that is with the portion of it which is directly available for the movement of the balance. We must, then, deduct from the entire motive force the portion that is lost by resolution.

This will be made clearer by an example: Consider a cylinder escapement with curved inclines of slight elevation; if the force of the motor is doubled we shall considerably increase the energy of the drop without doubling the energy of the impulse; so that, if the weight of the balance were doubled, the angular movement would not be in proportion to the increase in the motive force.

1341.—In any watch considered by itself, with all the depths remaining unchanged, we may take as a starting point the impulse obtained from any particular mainspring in order to find the proportion between the weight of balance and the impelling force, but this is not permissible when different watches are compared, because the precise amount of energy absorbed in the train varies from one watch to another, especially if the callipers differ. Hence we cannot regard as exact the table of sizes and weights of balances that C. Frodsham deduced from the cubical capacities of the barrels; it can only afford approximations that are more or less subject to error.

An example will make this point clearer; we are indebted for it to M. C. E. Jacot, a clever Swiss watchmaker.

He resolved to devote very special care to securing the best possible proportions for his depths, prepared elaborate

tables for this purpose, and he was then able to detect a marked increase in the force acting on the balance in watches of the same external dimensions and made by the same manufacturers of movements; the mainsprings employed by other makers were generally too strong for watches of his own construction.

The workmanship in the factories of the town of Geneva is usually more careful than elsewhere; and thus their balances are for the most part slightly heavier in comparison.

1342.—The best distribution of the total weight of a balance between its several parts can only be determined upon experimentally, because the rigidity of the arms, which should not be thicker than is absolutely necessary, depends on the metal employed. With a view to ascertain what should be aimed at, we have taken the dimensions of a large number of balances, especially of those belonging to watches with good rates. We then cut them to pieces and weighed separately the rim, the arms and the central disc.

This operation, which led us to prepare a table of the mean sizes and weights of balances in actual use, gave the following result which, we think, has a certain importance for horology:

The best proportion to adopt between the weights of the several parts of the balance is as follows:

Ten-twelfths of the weight in the rim.

Two-twelfths „ „ arms and centre (**354**).

Observation.—The subdivision of the total weight as here indicated is deduced from an examination of a number of watches with good rates; it is a limit fixed by the rigidity of the metal of which the arms are formed, usually brass. In compensating balances, the steel arms can be made to carry a relatively greater weight. Thus in a chronometer balance having a total weight of 4·2 grammes, the arms and centre can be reduced to a weight of 0·5 gramme. The rim and compensating weights will then have ten-and-a-half-twelfths of the entire weight. Or the proportion will be:

In the rim	21
In the arms and centre ...	3
Total ...	24

Even using these arms the weight of the rim might be increased, but it is important to avoid all risk of vibration through the centrifugal action of the weights; for this would introduce a source of irregularity very difficult of detection.

EXPERIMENTAL DATA.

Tables of the weights and sizes of balances in general use.

1343.—We have taken from a number of well-made watches of average size the weights and diameters of the balances that gave rates sufficiently uniform for ordinary purposes, and have thus compiled the following

TABLE OF THE WEIGHTS
AND MEAN SIZES OF ORDINARY BRASS BALANCES
EMPLOYED IN MODERN WATCHES.

Diameter.	Difference between successive Diameters.	Weight.	Mean difference between successive Weights.
14 millimetres		19 centigrammes	
	1 millimetre		2 centigrammes
15 „		21 „	
	1 „		3 „
16 „		24 „	
	1 „		4 „
17 „		28 „	
	1 „		5 „
18 „		33 „	
	1 „		6 „
19 „		39 „	
	1 „		7 „
20 „		46 „	
	1 „		8 „
21 „		54 „	
	1 „		9 „
22 „		63 „	

IMPORTANT OBSERVATIONS.

1344.—It will be seen from this table that the motive force increases regularly (in proportion to the weight) but at a much more rapid rate than the diameter; whence it follows that, assuming everything to remain proportional, if we take the barrel cover as a basis of measurement the force will be deficient in the larger sized watches and it will be found necessary to replace the balance by one that is lighter. We thus have further evidence that the barrel cannot be taken as a measure of the balance.

We had hoped to give details as to the strength of mainsprings, but the information we have obtained from a number of fellow-workers to whom we applied has been so incomplete and often contradictory, that it was impossible to confirm or

control the results of calculation by a sufficient number of experimental data.

By taking as a starting point a watch that is known to have had a satisfactory rate for some time past, with a balance beating 18,000 vibrations per hour, and strictly comparable with one of those given in the table, any watchmaker can calculate for himself the strength of mainspring that would be required with any other size of balance; by this means he will secure an approximation that will be excellent as a guide.

Or he can commence with a comparison watch such as the one whose dimensions are given in paragraph **441.**

In all the balances enumerated in the above table the total weight was distributed in accordance with the conditions laid down in article **1342.**

1345.—The manufacturer should bring himself to see clearly that he will never be anything more than a very average copyist unless he is competent to determine for himself the relation between the weight of a balance and the motive force that is designed to maintain its movement; for he must not forget the considerations we have already urged, namely that the motive force cannot be accurately represented by the strength of the mainspring, because, besides there being sources of error in the mechanism by which it is transmitted, the energy of the impulse that maintains the movement of the balance varies according to the manner in which the force that is transmitted by the lever or radius of the escape-wheel is resolved, as well as with the period occupied by the lift. It is, then, important not to leave out of account any one of these considerations in determining the weight of a balance that is to serve as a pattern for a particular calliper of watch.

1346.—As compensation balances are not used except in rather large watches, their weight is necessarily greater than those given in our table, but they are always proportioned to the increase of motive force. Hence the table will serve as a guide for them.

A clever watchmaker, M. Martens, has published in Germany some tables of the sizes and weights of balances; they only in part accord with our own determinations and appear to apply to movements made at Chaux-de-Fonds, or its district. Of these we give a selection in the following table; they may prove of service in furnishing a first approximation.

SIZES AND WEIGHTS OF CHRONOMETER BALANCES

1347. TAKEN FROM TIMEKEEPERS THAT POSSESSED GOOD RATES.

MAKER.	Weights in grammes including the compensating weights.	Diameter of rim in Millimetres.	Vibrations per Second.	REMARKS.
LARGE POCKET CHRONOMETERS, CHRONOGRAPHS.				
Jurgensen, U...	unknown	18.0	5	Subsequently incrsd. his bal.
Breguet	1.9 grm.	18.7	5	Employed various proportns.
Gannery.....	2.0 „	21.0	5	Had excellent rates.
Jacob	2.0 „	19.0	5	Ditto.
Cope	unknown	18.0	5	.....
French	do.	19.0	5	.....
Dumas	2.0 grm.	21.0	5	Had excellent rates.
Do.	3.5 „	22.0	4	.....
MARINE CHRONOMETERS.				
Berthoud, L. ...	3.5	27.0	5	Prismatic weights.
Do.	4.5	27.0	4	Weights projecting outwrds.
Parkinson	unknown	25.5	4	Provided with extrnl. screws.
and Frodsham	9.0	31.7	4	.....
Breguet	4.2	28.0	4	Employed very various prop.
Jurgensen, U...	unknown	27.5	4	Arc of vibration 450°.
Winnerl	7.2	27.3	4	Rnd.wgts. (anothr. dia. 27.7)
Gannery	4.5	27.0	4	Arc of vibration about 400°.
French	unknown	30.5	4	Two-day chronometer.
H. Robert	3.5	27.2	4	Arc of vibration over 500°.
Rodanet	7.2	27.2	4	Do., do., 350° to 360°.
Dumas	6.0	27.0	4	Round weights.
Leroy (Th.) ...	to	to		Mean arc of vibration about 400°.
Lecocq	7.0	28.0		
T. Adams	9.0	33.0	4	Rate well maintained.
Porthouse	7.0	29.5	4	.....
J. Poole	10.0	31.1	4	Round weights.
Mercer	unknown	30.5	4	Do.
Ch. Frodsham	9.0	31.0	4	Weight given is apprximat.
R. Roskel	unknown	28.7	4	Heavy weights.
P. Le Roy	152.5	121.7	2	Arc of vibration 120°.

To this second list we would add the names of MM. Jacob, Vissière and L. A. Berthoud, who have all adopted for 4 vibrations to a second a mean diameter and weight of 27 millimetres and 6 or 7 grammes respectively.

1348.—The English have adopted two principal sizes or patterns of chronometer. Balances measuring 30 millimetres and over are employed in the larger pattern. The other approximates very closely to that adopted by French makers.

The English usually take a complete circle or slightly more as the maximum extent of a vibration.

Several of the chronometer-makers that are here quoted have experimented with proportions that differ widely from any of those given in the tables. Nearly all have finally reverted to dimensions that were included within these limits.

1349.—An examination of these two tables brings prominently into view the relation between the annular balance and pendulum to which we have already directed attention (**1328**), and it shows that our theory rests on a sound basis.

The results we first gave (**1332**) fix accurately the mean proportion at which all the chronometer-makers have arrived, but this was only after a vast number of trials, many of them fruitless. When they deviated too much in either direction from the limit we have indicated, the rating of their instruments became tedious, variable and often impossible.

The balance of the earliest chronometer that was made on the principles that are in effect admitted to be true by all the above makers, that by P. Le Roy, does, if we make due allowance for the resistance opposed by the suspension-spring, come within the limits we have laid down as regards the radius of gyration. This coincidence is remarkable.

The differences, although slight, that these tables prove to exist between the balances of various makers, who have all successfully employed the dimensions there given, are probably still less than the figures would appear to indicate.

Indeed, as a general rule, the large balances of the English makers, with heavy compensation weights, are more charged at the axis and thus give rise to a considerable amount of friction at the pivots; and, comparing them with French balances, this increases still further the difference between the actual radius and the radius of gyration.

Had it been possible to determine with accuracy each radius of gyration, and had we possessed exact knowledge as to the amount of the retardation due to friction (**1336**), we feel satisfied that it would have been possible to deduce from these tables the *mathematical law* that determines the dimensions of the balance for any given movement.

Having laid down the principles on which this determination must be based, we have only one thing to regret; namely, that want of assistance or of time has prevented us from more nearly completing the work, which, however, is already considerably advanced.

PROPORTIONS OF COMPENSATING BALANCES FOR WATCHES.

As given by M. Martens.

1350.—As we have already observed, the proportions given by this horologist appear to be taken principally from timekeepers made in the canton of Neuchâtel, which has produced callipers of very great variety. Hence we see that the increase in weight corresponding to a given increase in diameter is irregular, not only when comparing the several tables, but even in the same table.

The diameters were taken from the barrel covers (**1323**) and the height of the balance is about four-ninths of the width of the mainspring. From these facts we can deduce the callipers that were taken by M. Martens as typical.

We are only given the total diameter inclusive of the projection of the screws. It is therefore impossible to deduce the radius of gyration, since the distribution of the mass is not indicated.

We give these tables then merely as practical data, sufficient to afford a first approximation. (The units of measure and weight are the millimetre and gramme.)

LEVER AND DUPLEX WATCHES.

Diameter.	Weight.
14 mm.	0·26 grm.
15 „	0·32 „
16 „	0·37 „
17 „	0·42 „
18 „	0·50 „
19 „	0·58 „
20 „	0·66 „
21 „	0·74 „
22 „	0·82 „

POCKET CHRONOMETERS.

Diameter.	Weight.
16 mm.	0·53 grm.
17 „	0·58 „
18 „	0·64 „
19 „	0·69 „
20 „	0·74 „
21 „	0·82 „
22 „	0·90 „

LARGE POCKET CHRONOMETERS.

Diameter.	Weight.
16 mm.	0·69 grm.
17 „	0·74 „
18 „	0·80 „
19 „	0·87 „
20 „	0·95 „
21 „	1·03 „
22 „	1·08 „

Concluding Notes.

Weight and velocity; which must be regarded as an element of timing. On the equilibrium of the balance.

1351.—It is said by some that we may replace weight by velocity and conversely in discussing the efficiency of moderators. This statement is only in part exact for the case of an increase or diminution in the number of vibrations; thus, for example, if we double the thickness of the rim of a balance, the radius of gyration and therefore the circumference of gyration will increase, so that we shall at the same time increase the weight and velocity; and by diminishing the rim we should have produced a contrary effect.

1352.—In stationary clocks weight is the main element of timing.

In portable timekeepers it is better to increase the velocity. For:

(1) In virtue of the laws of inertia, as we have already seen, a heavy body that rotates slowly is easily affected by any resistances that oppose its movement, a shaking of the supports, etc.

(2) The friction of the pivots is proportional to their diameter, according to the experiments of Berthoud and Romilly (and we are not aware of any that are more accurate), and the former states that it increases with the product obtained by multiplying the velocity into the mass, that is the momentum, or approximately so. It will, then, differ all the more in the vertical and horizontal positions according as the balance is heavier. (See articles **423** and **1439**.)

We may mention, as an example, an eight-day watch by Romilly, with a balance 26 mm. in diameter, which gave one vibration in a second and weighed about one gramme; according to a report presented to the French Academy it possessed an excellent rate when maintained in one position but went very badly when this was varied from the vertical to the horizontal, etc.

This example shows that a study of the rate in the several positions forms an element in the determination of the weight of a balance.

We shall revert to this subject in the articles on Timing.

1353.—A balance with uncut bi-metallic rim tends to become oval on a change of temperature. It would be much preferable if the bad bi-metallic balances employed by many manufacturers were replaced by balances with a heavy rim of one metal carrying screws.

1354.—The equilibrium of the balance may differ in vibrations of varying extent, owing to the fact that the pivots or pivot-holes are not perfectly round or the centre of gravity of the balance-spring is not stationary. Such a fault is especially liable to occur in pocket chronometers.

Accidental or permanent changes in the equilibrium of a compensating balance prevent the two weights from producing identical effects; one will have a greater influence than the other according as the position of the chronometer in a vertical plane places the excess weight up or down. There may moreover be a disturbance of the isochronism.

INFLUENCE OF CENTRIFUGAL FORCE ON THE COMPENSATING BALANCE.

1355.—The tendency of a rotating mass to fly from its

centre of rotation *increases as the square of the distance between this centre and the centre of the mass.*

If this distance remains the same while the velocity varies, the centrifugal force increases as the square of the velocity **(122)**.

In a chronometer balance, then, the centrifugal tendency increases as the compensating weights are made heavier and the arc of vibration increased.

If the semi-circular bi-metallic arcs do not offer a sufficient resistance, two sources of error may arise:

(1) The weights will cause the moment of inertia of the balance to alter when they fly apart.

(2) The segments of the rims will have a continuous vibratory movement, or this may only occur during the long arcs, the rim offering sufficient resistance in the short arcs.

It is unnecessary to do more than draw attention to these sources of irregularity.

We have on several occasions detected this trembling movement and tendency of the weights to fly apart by placing a fixed metallic arc in close proximity to them. The weights only touched it towards the end of the longest vibrations. We shall revert to this question in the chapter on Timing.

INFLUENCE OF TEMPERATURE ON THE COMPENSATING BALANCE.

Loss at extreme temperatures.

1356.—It is generally accepted as a fact (since any differences that there may be are inappreciable by our methods of experiment) that the dilatation of metals takes place uniformly as the temperature is increased from 0° C. to 100° C.

In a series of experiments on chronometers, Dent found that the tension of the balance-spring varies almost exactly with the temperature.

The experiments of M. Rodanet of Rochefort, described in the *Revue Chronométrique*, also show that the variations in the force of a balance-spring follow the changes of the thermometer, and they point to the conclusion that the compensating weights move along a secant to the circumference of the balance, the amount of movement towards the centre gradually becoming less as the temperature is more elevated; and therefore increasing in the opposite direction.

In order that the amount of movement towards the centre might be proportional to the change of temperature it would be

necessary that this secant should become a radius; but that is impossible with the ordinary balance.

Let us assume that by some suitable arrangement we have realized this condition and thus have a rectilineal compensation in which the movement takes place exactly as the temperature. Will the balance be thereby improved? We proceed to examine this question.

1357.—For a long time chronometer-makers have been aware of the fact that if a chronometer is accurately timed at a mean temperature, 15° C. for example, it will lose at the two extremes 0° and 30° C., but Dent was the first to publish this fact and offer an explanation of it.

The elastic power of the balance-spring varies inversely with the temperature and, in order that the rate of the chronometer may remain the same, it would be necessary that the effect of its compensation weights should follow the same arithmetical progression as the thermometer; but this is not the case, since the moment of inertia of the balance varies with the square of its radius of gyration, and therefore this latter diminishes in heat and increases in cold in a proportion that is altogether different from that of the thermometer.

In other words, in order to make our meaning understood by those who are not accustomed to the language of mathematics, assume that a radius of gyration of 10 mm. increases 1 mm. for x degrees fall of temperature, and that the force of the balance-spring increases with each x degrees fall in proportion to the numbers,

5, 10, 15, etc.

or by an equal amount of rise for each degree fall of temperature; the moment of inertia of the balance is represented by the figures

10^2, 11^2, 12^2, 13^2, etc.

or 100, 121, 144, 169, etc.

or there is an increase between the successive positions of

21, 23, 25, etc.

so that the increase becomes gradually greater in amount.

Hence it follows that the moment of inertia increases more rapidly than the force of the balance-spring and therefore there is a loss.

We might demonstrate in a similar manner that a chronometer timed for a given temperature will lose in heat; and that if timed for two extreme temperatures it will gain in intermediate temperatures.

1358.—A compensation balance then even if it be rectilineal will still be defective, for in neither case will the variations of its moment of inertia follow the same law as do those of the force of the balance-spring or of the other causes of gain or loss whether due to the train or the escapement.

Auxiliary Compensation.

1359.—With a view to avoid the irregularities that occur at extreme temperatures a vast number of forms of balances or of appendages to the ordinary balance have been suggested. They may be subdivided into two classes.

In one class additional bi-metallic strips are adapted to the balance so as to occasion the requisite increase or decrease in the radius of gyration at the extreme temperatures.

In the other class pieces are so placed that they do not move until the bi-metallic strips of the balance come in contact with them and their weight is thus added to that of the compensation weights, etc.

These additions necessarily complicate the balance, and, if not perfectly made, will add their errors to those of the balance itself. And besides this it has long been known that pressures and contacts, which from the nature of the case are variable, cannot be safely relied upon to produce the delicate and almost inappreciable effects of compensation.

Moreover the want of success of a great number of auxiliary compensations is easily explained when we remember the amount of care required and the difficulty involved in the construction and adjustment of a good balance, one in which the effects of the two arms are absolutely identical and coincident; and it is manifest that a complexity in the arrangement and a multiplicity of actions will increase the difficulty of making and adjusting the balance; indeed they will render absolutely essential both perfect workmanship and metals whose purity and homogeneity are quite exceptional.

The number of auxiliary compensations that has been invented is very great; we cannot attempt to discuss them. And yet some possess real interest, among which we would class one that is due to M. Vissière, a skilful chronometer-maker, and this we will proceed to describe.

Ordinary compensating balance provided with an auxiliary compensation by M. Vissière.

1360.—This balance, which is shown in fig. 6, plate XXI.,

is described by its inventor as a balance provided with compensation weights on supplementary bi-metallic strips.

A. An ordinary compensating balance.

a. A slide that moves on the balance rim and is provided with a clamping screw.

b c d, a supplementary circular bi-metallic band provided with an arm *e*; it is divided at *g*.

f, a screw that is perforated and tapped internally, serving to secure the strip *b c d* to the slide *a*.

h, a heavy mass set on the strip *b c d* near the point at which it is divided, and fixed by a screw *i*. At the centre it carries a screw *n* whereby its weight can be modified.

k, a screw for setting the whole in equipoise.

l, the ordinary compensation weight carried on the rim of the balance, whose weight is so proportioned to that of *h* as to secure the requisite compensating effect.

1361.—*Note by M. Vissière.*—"An auxiliary compensation should not merely secure a uniform rate at extreme and mean temperatures, but it should also maintain this uniformity in passing through all the intermediate temperatures and should be effective through a greater range than 0° to 30° C; these being the limits usually adopted (*see* the articles on Timing of Chronometers).

"The weight carried on a supplementary bi-metallic strip, *b c d*, possesses these properties. The strip has brass outside and steel inside.

"The weight *h* is so adjusted as to be on the prolongation of a radius of the balance passing through the centre of the circle *b c d* at the temperature of + 15° C. If the temperature rises or falls the weight *h* will move, through the dilatation of *b c d*, to one side or another, approximately along the circumference of this strip; but, since it is not concentric with the balance, it follows that the weight *h* will approach the centre on either side of this radius and will tend to accelerate the rate of movement of the balance. This acceleration will be the greater according as the weight is displaced further from its normal position.

"It is possible to adjust the compensation to within 1 second for a change of temperature of 50° C.

"I have not yet tested the system beyond this point, but have reason to believe that it will be possible to employ the system even for a greater range."

On the acceleration observed in the rates of chronometers.

1362.—Even the best chronometers are usually observed to gradually accelerate in their rate, after going for two or three years, by about 4 or 5 seconds per day, an amount which is of considerable importance.

Dent attributed this acceleration to the combination of oxygen of the air with the steel balance-spring, so that after a time its rigidity is increased.

M. H. Robert did not admit this explanation but considered that the gaining rate mainly arises from the fact that the resistance opposed by oil at the pivots of the escape-wheel differs from that at the pivots of the balance.

M. Jacob suggests the following explanation:

Chronometers are exposed to heat oftener and for longer periods than to cold, and since the balance is thus more frequently contracted, it follows that after a time the strips will not return exactly to their initial positions; there will therefore be necessarily a slight acceleration of the rate.

Lastly M. Villarceau attributes this gaining rate to the influence of the escapement, and he considers that it arises from the fact that the impact communicating the impulse occurs before the balance has arrived at its neutral position (**1364**).

MAXIMUM EFFECT OF A BALANCE.

1363.—A watchmaker, M. Mousquet, has endeavoured to ascertain whether between the longest and shortest possible arcs of vibration, there is any one particular arc at which the work performed by the balance is a maximum.

He has found that by gradually increasing the motive force of a chronometer so that the arc of vibration, commencing at 135°, becomes 490°, the *maximum effect* (which depends on the relation subsisting between the mass, diameter and arc of vibration of the balance and the motive force) is produced between arcs of 340° and 370°.

This result is very remarkable, for, as we shall see in the article on Timing, this is the mean arc of vibration adopted by very many chronometer-makers at the present day.

The question as to which arc of vibration is most likely to secure regularity is complicated; for this reason we cannot enter on a discussion of the problem suggested by M. Mousquet, but we commend it to the notice of watchmakers, more especially

since it is also applicable to pendulums (see *Revue Chronométrique*, Vol. II., page 212).

THEORY OF THE COMPENSATION BALANCE.

1364.—A physicist, M. Villarceau, has published, in the *Annales de l'Observatoire de Paris*, a theory of the compensation balance. This abstract mathematical research is beyond the reach of watchmakers, who alone could experimentally verify its conclusions and practically apply the consequences deduced by the author from his calculations.

TO MAKE A COMPENSATION BALANCE.

1365.—Cut from a carefully selected bar of cast steel a disc somewhat larger and thicker than the finished balance. Some makers use worn files that have been softened for this purpose.

After trimming the disc a hole is drilled at its centre in the uprighting tool of the same diameter as the balance-staff. This hole is broached with the greatest possible care and, when the disc has been turned flat on one face on a smooth arbor, it is fixed with shellac on a chuck in the lathe. This chuck is provided at its centre with a short arbor, hardly as long as the disc is thick, which fits the hole in this latter without play and serves to centre it.

Those unaccustomed to the operation of turning the disc may strain the arbor and sometimes loosen the shellac. If accurate work is looked for, it will in such cases be necessary to replace the arbor whenever the disc is re-fixed; but when some experience has been acquired this will be found unnecessary.

The edge of the disc is turned perfectly square and the diameter reduced to that of the required balance less the thickness of the external ring of brass (from two-thirds to three-fifths the total thickness of the rim).

The disc is now removed and the central hole plugged. Some makers employ a turned stick of slate, accurately fitted in the hole, and then cut off and slightly bruised on either surface.

Others fit in a rod of highly burnished steel. One recommends that it be first passed through an onion, as thus a thin layer is formed on its surface that prevents adhesion; most makers, however, prefer to merely dip it in the oily matter on an oilstone. It is then cut off flush with either side. It is essential that the hole be hermetically closed because, if the

slightest particle of brass enter, the two pieces of steel will braze together. The disc may be covered over on either side with white-lead made into a paste with water.

The disc is placed in a small crucible from $1\frac{1}{2}$ to 2 inches wide and from 1 to $1\frac{1}{2}$ inches deep, the side that was waxed to the chuck lying flat on the bottom of the crucible, and is covered with the best quality brass, the amount of which varies from one-third to one ounce according to the size of balance, and the whole is freely covered with finely-powdered borax.

The edge to which the brass is required to adhere must not have been touched by the fingers and, in order to ensure the brazing, it should be previously painted over with a paste of borax and water.

The crucible is now placed in the muffle of a cupelling or reverberatory furnace as the draught can then be regulated, which is not the case in an ordinary forge fire; previously the furnace should have been raised to a red heat. After a few minutes the fusion of the brass will commence. Shake the crucible, holding it in a pair of tongs, or stir the molten metal with an iron rod in order to bring all impurities to the surface. The fusion will be complete when the vapour of zinc is distinctly seen to rise. Withdraw the crucible and, by means of the iron rod, force the steel, which floats, to the bottom of the crucible and maintain it there until the brass is partially solidified.

If the heating is excessive or not uniform it will give rise to blisters, etc., which will either be found during the subsequent turning or remain concealed within the thickness of the brass rim and cause the compensating action to be irregular.

The surface of the steel that was in contact with the crucible will have very little brass adhering to it; this must be removed and the steel smoothed with care. The brass on the other surface is removed with a file as well as the greater portion of that round the edge, only reserving a rim about 2 or 3 mm. (0·1 inch) thick.

The central hole is next cleared by forcing out the pin that filled it; if the requisite precautions have been taken, the hole will be found to be perfectly clean, and it will only be necessary to polish it with a stick and rouge.

1366.—The smoothed face of the disc is fixed with shellac on the chuck with a central arbor, after this has been carefully tested to ensure its absolute truth. The exposed face is turned down and the balance may be reduced to its final thickness.

Then the brass is turned perfectly square, leaving its thickness somewhat greater than is required, and it is reduced to very nearly the exact thickness by hammering, which process is conducted with very great care so that the metal may be of uniform hardness throughout.

Formerly it was hammered on its rim after being removed from the lathe. At the present day, however, many manufacturers prefer to harden the brass while the balance is mounted in the lathe by means of a milling tool with fine notches. They repeat the operation several times after having removed the notches produced and are thus enabled to reduce the brass to the required thickness without having to detach from the chuck.

If the hardening be continued for too long the rim will close in when it is divided. If it be carelessly performed or uneven one of the arms may close in further than the other or rise above the flat of the balance which then becomes worthless.

The rough surface is removed from the brass and the inside cut away to the depth previously determined upon, either by hand or by using a pointed cutter in the slide-rest; and the surface should be finished with a flat ended cutter but the width of the acting edge must not be more than 0·2 mm. (0·008 inch). These cutters should always be very carefully set, and should remove the metal in long thin threads; for otherwise they will scrape the surface and cause it to cockle, an effect which is very detrimental as it resembles the action of rolling.

The interior and the flat of the rim are smoothed with fine emery powder, the angles being slightly rounded off.

After the balance has been removed from the chuck and the shellac cleared from its under surface, the holes for the screws are drilled in a small drilling machine arranged for this purpose. These holes (numbering twenty or thirty when the balance is only provided with screws and does not carry weights) are set gradually nearer together towards the free ends of the strips. The finer they are the better.

It is best to tap them in a tool specially prepared for the purpose.

The two arms are next crossed out; they should be perfectly uniform and alike in shape so as not to disturb the equipoise of the balance by being of unequal weights. This operation requires some care and lightness of touch for it is necessary to avoid touching the interior of the rim.

The surfaces are now smoothed and the file marks removed by drawing an iron charged with oilstone dust across them.

The compensating weights are next adjusted.

If the slot is straight it can be made with a circular cutter in the wheel-cutting engine; but it is better to give it the same curvature as the bi-metallic rim. This may be done in several ways; thus a groove may be cut in a thick brass plate, either by a single cutter fixed in the slide-rest or by a series of cutters fitted together so as to form a kind of circular saw with the edge projecting from the flat and of the same thickness and diameter as the bi-metallic rim; and the weights subsequently cut from this plate.

One of our best-known French chronometer-makers, M. Rodanet, merely turns the weights to the required form, and then fixes them with wax in a deep circular groove cut in a thick plate that is just large enough to admit them. If the plate be now centred in the lathe, slots can be cut in all the weights at once.

When the weights and screws (which should fit firmly with friction) have been completed, the two segments of the rim are cut through with a rotating cutter in the wheel-cutting engine. It is well, as a precaution, to fix the rim of the balance with wax on a plate that has two openings to admit the cutter. The balance had better now be placed in oil, which is caused to boil so as to bring the whole to a more uniform molecular condition, as the metal is always somewhat distorted in the course of the work. The balance is next cleaned and fitted with screws and weights; when it has been fixed on the staff and poised on the pivots it will be ready for adjustment.

Observations.— The lathe employed should be very well made, perfectly adjusted and very solid. The arbors, collets, etc., in short all the fittings should preferably be short and solid, and their size should be such as to prevent vibrations.

The several slides should have a bearing of sufficient breadth, and the screws should always act regularly and without jerks or backlash.

To make an ordinary balance
with uncut rim.

Our small practical manual (The Watchmaker's Handbook) gives all necessary details on this subject; we therefore refer the reader to it.

THE BALANCE-SPRING.

Historical Notice.

1367.—We have had in our possession a watch that was apparently made in the very earliest days of portable timekeepers. The balance was a *folliot* (**141**) and carried on its axis a thin straight flexible blade that struck against two pins fixed in the plate at each movement of the folliot and the blade was thus more or less deflected.

In old timekeepers, usually of German manufacture, the following arrangement is occasionally met with; two pieces of silk or pig's bristles are fixed to the plate and project above or below the rim of the balance. At each movement of this latter a pin set in its rim causes the two bristles to be alternately deflected and the balance oscillates from the one to the other as though driven by their elastic force.

Must we regard these primitive devices as the origin of the balance-spring? Nothing justifies us in doing so.

A watch provided with a balance-spring, very nearly of the form used at the present day, appeared in London in 1675, having been constructed under Huyghens' directions, "and," says Derham, "it excited as much interest as if a method had been discovered of determining the longitude at sea."

It is well known that if an elastic lamina be fixed at one end and the free end be deflected from its position of rest, it will perform oscillations that are practically equal on either side of this position for a certain period. From this property of elastic blades it resulted that when a spiral spring was attached to a balance the regularity of its movement was relatively so great that the invention of Huyghens was at once regarded as of the highest importance. Hence the credit of the discovery was eagerly claimed by Hautefeuille and Hooke.

The dispute may be briefly summarized as follows.

Hautefeuille was the first to publish (in 1674) the fact that a spring applied to a balance would facilitate its oscillations and increase their regularity, but he does not seem to have had any idea of fixing it to the axis of the balance itself or of giving it a spiral form.

He moreover proposed, according to Moinet, to bend a straight spring in waves. But Romilly attributes this idea to Lahire. There is unquestionably some confusion and this wavy

spring is probably nothing more than the helical spring of Hautefeuille.

Huyghens took up and carried out the novel idea of Hautefeuille, employing a spring coiled up into a spiral. The success of his experiments caused Hooke to claim priority.

It appears certain that the latter had an idea of employing a straight spring as early as 1660, and a few years later it is said that he actually applied a spiral spring to a watch, but it was much shorter than the spring now in use. Hooke's invention however was maintained secret until the appearance of Huyghens' watch and it is therefore legitimate to conclude that he had not fully appreciated its importance.

1368.—HELICAL BALANCE-SPRING.—The ingenious Abbé Hautefeuille was moreover the first to employ a cylindrical or helical balance-spring in horology. But in this case also, doubtless owing to the want of some technical knowledge, he had the bad fortune not to perceive all the advantages of his system. Thus he only utilized the elasticity of his coiled spring in the direction of its axis and not at all in order to produce a circular motion by tension and distension round this axis. Fig. 9, plate XIX., is taken from his work.

The following very curious passage occurs in it:

"We can," he says, "not only employ a helical spring (as shown in fig. 9) but also springs having every conceivable form to be found in nature, providing that they are capable of performing vibrations: *we may perhaps be able to discover some that have special properties, such as causing the long and short arcs of vibration to occupy equal periods.* With this object in view I proposed the use of anisocyles, that is to say springs formed with successive coils of unequal diameter and coiled on a cone."

Were it not that the author's explanations clearly show that he only intended to employ the tension and elastic reaction of the spring along the axis of the cylinder or cone formed by the spring and not by the angular motion round this axis, one would be disposed to attribute to him the discovery of the isochronal balance-spring, but no confusion or error is admissible in such a claim.

1369.—ISOCHRONAL BALANCE-SPRING.—In 1766 Pierre Le Roy thus enunciated his discovery of the isochronal balance-spring:

"There is in every spring, providing it be long enough, *a*

length that causes all the vibrations, whether long or short, to be isochronal; having fixed upon this length if you *shorten* the spring, *the long vibrations will be quicker than the short;* if, on the other hand, you *lengthen* it, *the short arcs will occupy less time than the long.* It is on this important property of the spring, which has hitherto been unknown, that the regularity of my marine chronometer mainly depends." (It had a going barrel and no fusee.)

P. Le Roy employed two flat superposed springs opening and closing in the same direction (fig. 10, plate XIX.). Since the arrangement of his movement only allowed the very large balance, at that time considered essential, to perform short arcs of vibration, the two springs had, within the range through which they acted, the properties of a helical spring.

Ferdinand Berthoud in vain endeavoured to appropriate the beautiful discovery of P. Le Roy. At first he published a geometrical theory of it, from which he deduced the fact that we can secure isochronism of the oscillations of a balance by using a spring whose strength gradually diminishes from one extremity to the other in a certain definite progression, and he invented the tapered balance-spring.

P. Le Roy would not admit that isochronism should be aimed at by a curve adjusted by trial, such as Gourdain, a French maker of that day, proposed, prior to 1770, to adapt to the balance-spring with a view to accelerate the long arcs and thus make them equal to the short arcs. In this we perceive the origin of the various terminal curves for chronometer balance-springs employed at the present day; but we are bound to add that, if the proposal of Gourdain failed to find acceptance in France, this was doubtless due to the fact that in his day only flat springs were employed. His device, of which we have never been able to find a description, was probably somewhat analogous to the Breguet spring employed at the present day.

In 1776, J. Arnold used cylindrical springs in his chronometers, and in 1782 he took out a patent for a spring of this form in which the last coil was turned in towards the centre (fig. 11, plate XIX.). "These terminal curves," says the specification, "possess the property of rendering all the vibrations of equal duration, since the figure of the balance-spring always remains similar to itself."

Lastly M. Phillips of the French Institute has in recent

years published, in a very exhaustive memoir, the theory of the terminal curves of balance-springs that secure isochronism.

Various forms of balance-springs.

1370.—L. Berthoud employed conical springs with success, that is to say springs formed like a fusee (fig. 13, plate XIX.). He found them to possess a marked superiority over cylindrical balance-springs in regard to the progressive increase in the force exerted, and from the fact that the coils, although very near together, could not come in contact.

1371.—A. Breguet employed a balance-spring bent and lying in two planes, now generally known as a Breguet spring; only the external coil is bent into the upper plane (fig. 17, plate XIX.).

He also proposed to employ a spring that was thicker at the two extremities and gradually became thinner towards the middle. Such a spring assumes the form of a cask when it opens out and, on closing, resembles a spindle narrower at its centre than towards the extremities. The advantage secured by this device, which is also realized more easily by forming certain terminal curves at the extremities of the ordinary cylindrical spring, consists in the fact that the spring itself is not thrown on one side in the long arcs of vibration.

1372.—The *spherical* or globular balance-spring (fig. 16, plate XIX.) was first made by Frederick Houriet.* He asserts that this form enables the balance to perform the greatest possible arc of oscillation for a given impulse. If we compare this with a cylindrical spring having proper terminal curves, the advantage of the spherical form will be found to be either nothing or very slight (**1374**).

1373.—Finally attempts have recently been made in England to adopt the Hammersley balance-spring. The middle portion of the lamina is formed in a cylinder and the two extremities form flat spirals towards the axis, the two planes of these spirals being parallel to each other and perpendicular to the axis of the cylindrical portion. It is, in short, a triple spring, being formed of a cylindrical balance-spring terminating in two ordinary flat springs (fig. 19, plate XIX.). It has been

* F. Houriet was a Swiss watchmaker born about the middle of the last century. He worked for nine years in Paris with some of the best makers, P. Le Roy, Romilly and F. Berthoud, and returned to establish himself in Neuchâtel. The rapid progress made by the watchmakers of that canton was in part due to his efforts.

made with only a single flat portion; this is known as the double balance-spring.

1374.—This latter would certainly be preferable in watches of average thickness to the Breguet spring, but we cannot see what advantage the triple spring possesses in chronometers over the cylindrical spring with proper terminal curves. It does not in any way facilitate the movement of the balance and requires as much care as a spherical spring; and chronometer-makers have decided that the cylindrical spring with proper terminal curves is preferable to this latter for the following reasons: its construction is more simple, a hollow block can be used in the hardening, and it is possible to modify these terminal curves as required, thus materially facilitating the final adjustment in the timing.

It has been objected that the forming of these curves or their alteration with pliers would damage the metal or modify the molecular arrangement of the steel, giving rise to an accelerating effect that would gradually become less and disappear in time. But in order to avoid these influences with the Hammersley balance-spring or with the spherical spring it would be necessary to harden them when of their final form and never to have occasion to touch the extremities. This could only happen very rarely, since the tests of isochronism and in positions must of necessity be made after the chronometer is completed.

Springs in opposition and with double bends.

1375.—Romilly, about the end of last century, following out a suggestion of Bernoulli, applied to a marine chronometer by Frederick Houriet, who worked for him, two flat balance-springs one above the other and set so as to develope in opposite directions; each was coiled up once on itself so that the whole was in a state of constrained equilibrium. The freedom of movement of the balance was so great that there was some difficulty in stopping it.

P. Le Roy had already pointed out that there was a disadvantage in employing two springs so arranged because they are both in a constrained condition, and this he considered would give rise to sources of error greater than those that are avoided by reducing the lateral friction of the pivots, etc.

1376.—As early as 1838 a double cylindrical balance-spring was invented, and a description of it was published in a manual by L. Janvier and Magnier; it is formed of a single

band of steel bent double at its middle and the upper portion is bent into a spiral in one direction while the lower portion turns in the opposite direction (fig. 15, plate XIX.).

The intention of the inventor, who only succeeded in producing a balance-spring with two spirals in opposite directions but unconstrained, was not only to diminish the friction of the pivots, since one half of the spring closes to the same extent as the other opens, but more especially to ensure the resting position of the balance remaining invariable. In other words he hoped by this means to avoid the watch getting out of beat, which he considered possible when the balance-spring became longer at elevated temperatures (**1400**). This shows that these two horologists were somewhat ignorant of the laws of dilatation (**1259**).

1377.—More recently M. Rozé has devised a balance-spring of double curvature which however differs from those that precede in that the middle of the band (which is not bent) is at the centre of figure of the whole; each half of the spring is bent twice in opposite directions and then coiled into a cylinder, the coils of one half being right and the other left-handed. The force exerted by the ordinary cylindrical spring in the direction of its axis is thus avoided, a force whose influence is made very evident by an instrument constructed for M. Phillips by M. Rozé. The relative forces thus exerted when the balance is vertical and horizontal are, so far as we know, still undetermined.

SUMMARY OF P. LE ROY'S WORK ON THE BALANCE-SPRING.

1378.—When the balance-spring was first applied to a watch it was generally admitted that the vibrations of an elastic lamina, whether free or attached to a balance free to move on pivots, will all occupy the same period of time, although this is not exact (**1254**); just as they had assumed from the incomplete experiments of Galileo that the long and short oscillations of a pendulum were isochronal until Huyghens published the law that would enable us to ensure isochronism.

Since most watchmakers were without accurate means of verification and the uniformity in the rate of the new watches was relatively very good, they concluded, just as Hooke did, that the long and short arcs were isochronal; a few horologists, however, who were good observers and possessed a power of

analyzing effects, who moreover were engaged on the problem of determining the longitude at sea, disputed this conclusion.

In 1759, P. Le Roy published the following:

"From experiments that I have made it appears that the free vibrations of a balance with attached balance-spring *are not exactly isochronal.* If, for example, it performs 116 vibrations of 60° in one minute it will only perform 115 in the same time if the arcs measure 120°, etc."

This result is confirmed by those of Sully,* Gourdain, etc., and of Harrison, who had shown that the long arcs of vibration of his balance occupy a longer time than the short arcs (an effect which he remedied by his cycloidal stud).

1379.—Following up his earlier experimental researches, and relying on the mechanical principle that "the movements of a mobile are isochronal when the forces impelling it are proportional to the degree of tension," P. Le Roy made (prior to 1766) the following experiment:

"With a view to ascertain what conclusions I should draw on this important point (whether the force exerted by a spiral spring increases in proportion to the space traversed in expansions or contractions of varying extent) I took a large spring from an ordinary watch; its internal extremity was attached to a staff supported on very fine pivots which carried a large pulley; and the outer end of the spring was then made fast in such a position that it was in its neutral unconstrained state. I then attached a fine thread to the pulley, coiled it several times round and attached a light hook to the free end to which different weights could in succession be attached. As these weights strained the spring, opening or closing it to a greater extent than if it had caused a balance to vibrate, I observed the several distances through which the hook descended and found them always to be in proportion to the loads applied, etc."

We see from this extract that P. Le Roy was the first to suggest the *elastic balance* (**1424**) for measuring the force of balance-springs and their gradual divergence from true isochronism and that by its means he had found a spring that was really isochronal. What is so remarkable is that this great artist had foreseen, from these early experiments, that the isochronal spring

* An English watchmaker who established himself in France where he acquired considerable reputation. Dying in 1728, a friend of J. Le Roy, Sully contributed largely to the advance of the horological art in the country of his adoption.

would not behave in the same manner when working in conjunction with the escapement as when fixed to a detached balance, and that he convinced himself, by observation and reasoning, that a spring which is isochronal with the balance for measuring its strength may be no longer exactly so in a watch. This observation rendered his research complete and he was then able to formulate his great discovery in the terms given above (**1369**).

1380.—To that passage he added the following: "I am satisfied that if the shortest and the longest arcs are once rendered isochronal by this method, all intermediate arcs will be so also."

It is important to observe, as bearing on this last quotation, that the fact of this being the case under the special conditions that characterized P. Le Roy's chronometer does not justify us in concluding, as some watchmakers but little acquainted with practical chronometry have done, that the isochronism of long and short arcs necessarily involves the intermediate arcs being isochronal; this is generally not the case in ordinary modern chronometers.

This brief summary proves that the discovery of the isochronal chronometer balance-spring, was no more a happy chance than that of the detached escapement or the compensation balance; it resulted from the following up of an abstract idea by a man of genius who spared neither perseverance nor ingenuity.*

1381.—P. Le Roy moreover established experimentally, by means of the instrument that he termed an *elaterometer*, that the elastic force of a balance-spring diminishes as the temperature is increased, but that such a change does not alter the isochronism (when the arrangement is such as he adopted). He neutralized this weakening of the spring, as, following his example, we have since continued to do, by the movement of the compensating weights carried on the rim of the balance; whereas F. Berthoud resorted to that objectionable device, a movable pair of curb pins by which the acting length of the balance-spring was caused to vary.

1382.—It is further of interest to note that P. Le Roy

* Born in 1717 and dying without children in 1785, P. Le Roy, the son of Julien, is unquestionably the greatest horologist that has honoured and enriched France by his labours, and no foreign horologist, even the most celebrated, has done more valuable work. His labours are to be admired not merely on account of the value of the discoveries he made, but because during twenty years, without once losing sight of the main object, or resting until he felt satisfied the work was complete, he followed it out with the logic of real genius, the disinterestedness and the modesty of a noble spirit.

pointed out these two important facts: (1) That a spring loses a considerable fraction of its elastic force in the first few months it is in action; the weakening then becomes less, until at length it is almost insensible, unless the spring is overheated, in which case it cannot return exactly to its previous amount; (2) if the metal is not in a uniform molecular condition or if it is constrained it will give rise to variations.

He therefore recommends that the metal be always subjected to temperatures that are both higher and lower than those to which it will be subsequently exposed. He first fixed the internal extremity of his chronometer balance-spring, the other extremity being held in a free stud. This stud was not definitively adjusted until after the spring had been subjected to temperatures that were relatively high and low. In a memoir presented by F. Houriet to the Society of Arts of Geneva, several of these observations are recorded, but the writer omits to mention the source from whence he obtained them.

THEORETICAL CONSIDERATIONS.—DEFINITIONS, etc.

Moment and Coefficient of Elasticity.

1383.—The amount of a force (whether it be due to a mass, inertia or elasticity) is measured by the power exerted at its point of application at the moment under consideration.

The *moment* of a force round a given point is measured by the force applied multiplied into the virtual lever arm.

The *work done* by a force is obtained by multiplying the force into the space traversed by its point of application, measured in the direction in which the force acts.

When a body is moving under the action of a force (gravity, elasticity, etc.) which continues to act uniformly upon it, its energy is measured by multiplying the mass into the square of the velocity.

If the impulse, instead of being applied uniformly, is gradually increasing or decreasing, the energy will increase more rapidly in the first case and less so in the second than the squares of velocities would indicate. (See any treatise on Mechanics.)

In algebra a number that is set before any quantity, as a multiplier, is termed a *coefficient*.

If we study experimentally the elastic resistance offered by two rods, identical in form and transverse section but of different material, and if, under certain similar conditions, the resistances are 10 for the first and 15 for the second rod; 10

and 15 will be the coefficients of elasticity of the metals forming these rods. In any volume that treats of the strength of materials will be found tables giving the coefficients of elasticity of metals.

The *moment of elasticity*, represented by M, of a round rod of radius r and having a coefficient of elasticity E, can be calculated from the formula

$$M = E \times \frac{3{\cdot}1416 \times r^4}{4}$$

If the section of the rod is rectangular, the width (parallel to the axis of rotation) being a, and the thickness e, the moment of elasticity is given by the formula

$$M = E \times \frac{ae^3}{12}$$

The radius, in the case of a round rod, and the width and thickness when it is rectangular, must be expressed in fractions of a metre.

M. Phillips enters into very full details in his manual, and in his memoir he gives an experimental mode of determining the moment of elasticity of a spiral spring by employing a short piece of the lamina. As we cannot discuss this question further we must refer the reader to his two works.

Law of isochronism of the balance-spring.

1384.—We know that an isochronal balance-spring is one that possesses the property of causing the vibrations of different extent of a balance to take place in equal periods of time.

We further know that in order for it to have this property it is essential that the force exerted by it be always proportional to the angle of tension according as it is more or less coiled up. In other words if a balance-spring, fixed at its outer end, be attached by its inner extremity to an axis perpendicular to its plane that is movable on pivots, and if a force, represented by a gradually increasing weight, be applied tangentially to the axis, we shall know that the spring is isochronal when each equal increase in the weight (short of the point at which the metal would be permanently distorted) gives rise to an equal angular movement of the axis.

From the above explanation it will be seen that the limits between which it is possible to secure isochronism, of considerable extent when a cylindrical spring is employed, are much less so with a flat spring; and, further, that with a given balance the spring which is isochronal to it will be determined by a pro-

portion, to be ascertained experimentally, between the section and length of the metallic lamina, assuming it to be perfectly homogeneous and of equal section throughout its entire length.

A balance-spring that is too short, or, what comes to the same thing, is too thick in proportion to its length, will coil up too rapidly as compared with the weights applied. The converse is the case when the spring is too long.

Isochronism can be secured with greater facility and more perfectly by bending the extremities of a cylindrical spring towards the centre, adopting one of the curves whose theory has been given by M. Phillips of the Institute of Paris.

Terminal curves for the balance-spring.

1385.—In his memoir *Sur le spiral réglant*, M. Phillips shows that if the terminal curves of a cylindrical spring satisfy the following conditions (A B M C being the curve, G the centre of gravity of the curve, and O the centre of figure of the spring, fig. 74):

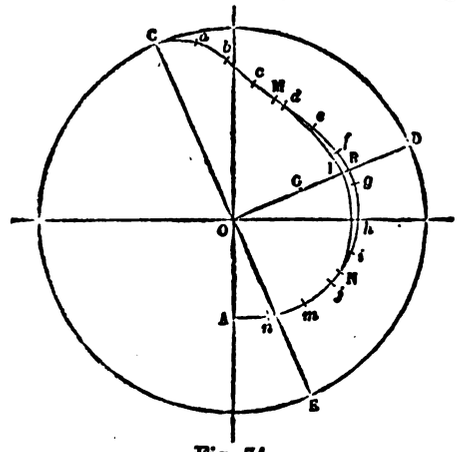

Fig. 74.

(1) O G must be perpendicular to the radius O C, C being the point at which the terminal curve A B M C starts from the body of the spring;

(2) O G must be a third proportional to the radius O C and the length A B M C of the curve, so that we have

$$O\,G = \frac{O\,C^2}{A\,B\,M\,O}$$

A balance-spring that is so arranged, the two ends being formed into parallel curves that satisfy these conditions, will possess the following properties:

(1) The centre of gravity of the spring will always be on the axis of the balance;

(2) The spring in opening and closing will always remain perfectly cylindrical and concentric with the axis, and its force will increase in proportion to the angle of rotation of the balance (principle of isochronism);

(3) The spring will not cause the balance, at any point of its movement, to exert any lateral pressure against the sides of the pivot-holes.

1386.—The author points out that calculation does not indicate that this is the only mode of securing isochronism. But it clearly proves it to be one method.

It is a matter of indifference whether the two curves be one over the other or not; they may be crossed or inclined at any angle.

Modes of graphically tracing out these terminal curves.

1387.—We would only repeat here the explanations given in M. Phillips' practical manual on the balance-spring, and, since that work can be obtained at little cost (2 francs), shall content ourselves by referring to it, merely giving, on the opposite page, a sheet representing a number of these theoretical terminal curves. Each is drawn in two sizes, the smaller being approximately the same as the springs ordinarily met with in practice.

It will be noticed that figure 21 is obtained by drawing two quarters of circles with a radius equal to one half that of the main coils and joining these quarters by a straight line. Figure 23 represents a half-ellipse. Its major axis is the diameter of the cylinder and the minor axis is 0·58 of the major. The length of this half-ellipse is exactly 0·8 of a half-circumference of the cylinder.

THEORETICAL AND PRACTICAL ISOCHRONISM.

1388.—Reverting to the principles already laid down, etc., it has been seen that, if we take a detached balance provided with a perfectly isochronal spring and adapt them to a chronometer, the isochronism in the vibrations of this balance that previously existed may be altered by: (1) The action of the escapement; (2) An appreciable difference in the size of the pivots; (3) A change in weight of the balance; (4) A sensible variation in the motive force.

In a word this isochronism will be more or less modified whenever the spring is subject to influences that interfere with that progressive increase or diminution in its force which is so essential to isochronism.

VARIOUS TERMINAL CURVES DEDUCED FROM THEORY.

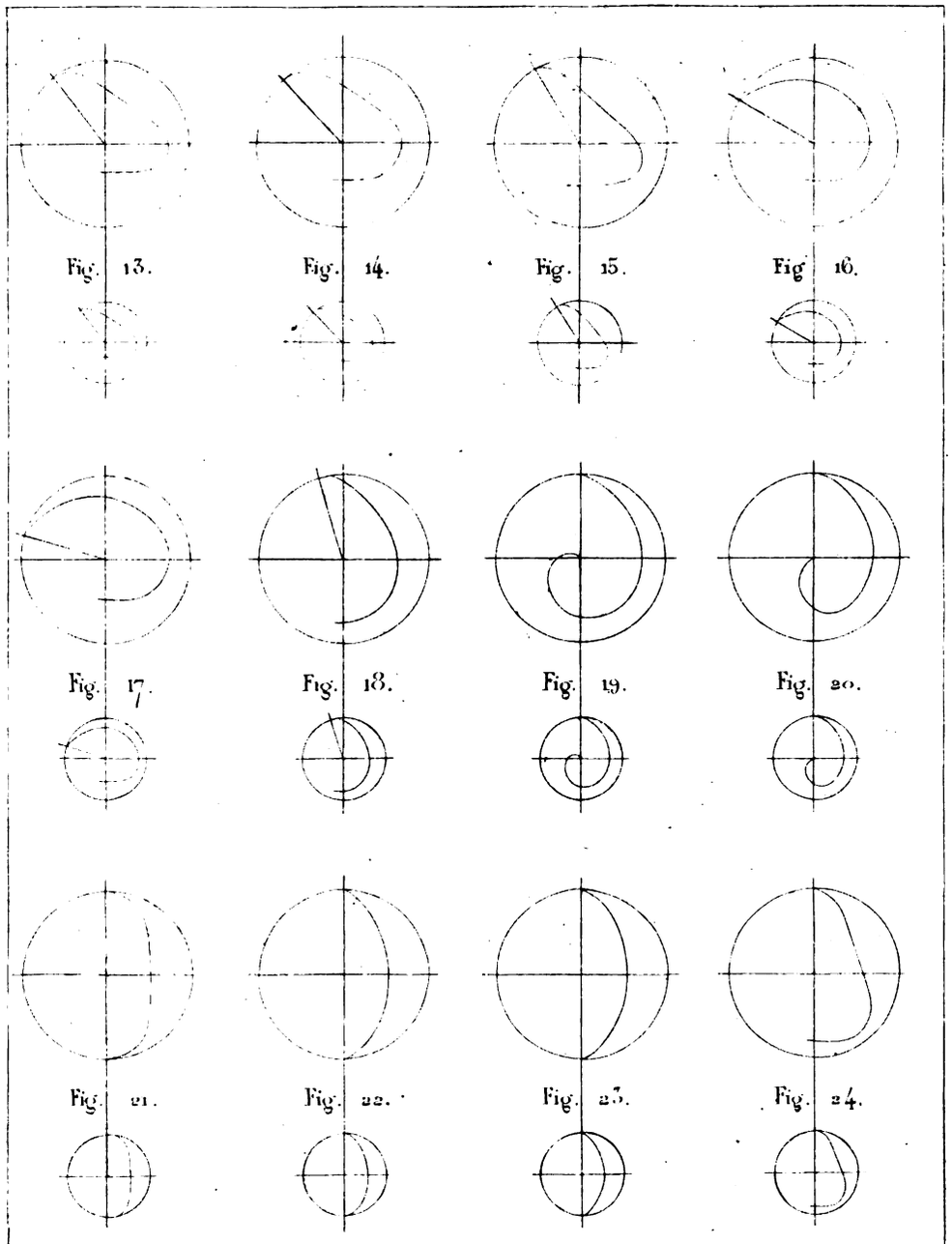

Fig. 13. Fig. 14. Fig. 15. Fig. 16.

Fig. 17. Fig. 18. Fig. 19. Fig. 20.

Fig. 21. Fig. 22. Fig. 23. Fig. 24.

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1389.—It is then important to distinguish between *theoretical* or *absolute* and *practical* isochronism.

Absolute isochronism is that which satisfies M. Phillips' theory of the balance-spring.

But this absolute isochronism cannot be realized in full in the chronometer because:

(1) We cannot guarantee the terminal curves, which of necessity are often slightly modified a number of times in the operation of timing, to be formed with such absolute accuracy as to be certain that all lateral pressure is neutralized and that the force of the spring varies progressively as required;

(2) The lift has a double action which is utterly destructive of absolute isochronism; it presses the pivots against the sides of the pivot-holes and changes more or less suddenly the character of movement of the balance (**1446**).

Practical isochronism, the only kind that can be attained to in a chronometer, is merely absolute isochronism modified in accordance with the changes introduced in the nature of movement of the balance:

(1) By the lifting action, which has a greater or less disturbing influence according as it takes place with greater or less force, for a longer or shorter period, etc.;

(2) By the variable resistances due to friction, etc.

1390.—A watchmaker that is desirous of producing high-class work will do well in the first instance to study the conditions that would require to be satisfied in order to secure theoretical isochronism. He will then be the better able to come as near as possible to it in practice because he can rely on a more thorough knowledge of the mode in which the escapement should act in order to secure practical isochronism.

Such a starting point appears to us to be of very great value and leads us to consider the publication of M. Phillips' remarkable memoir as of very great service to the horological art. If we possessed with an equal degree of certainty a theory giving the impulse at different ages of the oil, etc., we have no doubt that the science of constructing chronometers would be, what it is very far from now being, a positive science.

WORK DONE BY THE BALANCE-SPRING.

1391.—The amount of work performed by the metal constituting a balance-spring during its action is expressed by M. Phillips in a very simple formula, from which it appears that:

If the thickness of the spring is made twice, thrice, etc., as much, the amount of work done will be two, three, etc., times as much.

If the length of spring is doubled or trebled the metal will do two or three times less work.

If the balance turn through an angle that is greater, the work performed will be proportionately increased.

To calculate the period of vibration of a balance with attached balance-spring.

1392.—M. Phillips gives the following formula:

$$T = \pi \sqrt{\frac{A\,L}{M}}$$

where T is the period of a single vibration of the balance (taking the second as a unit).

A is the moment of inertia of the balance; M the moment of elasticity of the spring.

L the length of spring, taking the metre as unit of length; and π the ratio of the circumference to the diameter (3·14159).

1393.—The following law is involved in the above formula:

The number of vibrations (in a given time) *of a balance that is impelled by a spiral spring is inversely proportional to the square root of the acting length of this spring.*

In other words if two balances have the same moment of inertia and are moved by springs made from the same wires, the numbers of vibrations will be inversely proportional to the square root of the length of springs even although their diameters be different.

Practical applications.

1394.—*A balance performs a certain number of vibrations in a given time; find how much the acting length of the balance-spring must be shortened or lengthened in order that the balance may make some other number of vibrations in the same time.*

In accordance with article **1393** we have the proportion: $n : N :: \sqrt{L} : \sqrt{l}$.

The number of vibrations required (n) is to the initial number of vibrations (N) as the square root of the acting length (L) is to that of the required length (l).

Assume the balance to be losing 2 minutes in an hour, so that it makes 17,400 vibrations whereas it should make 18,000, and let the acting length of the balance-spring between its two fixed points be 194·8 mm.

The proportion will be:

$$18{,}000 : 17{,}400 :: \sqrt{194{\cdot}8} : \sqrt{l}$$

$$\text{whence } \sqrt{l} = \frac{13{\cdot}9 \times 17{,}400}{18{,}000} = 13{\cdot}5.$$

Squaring this quantity we obtain, for the length l, 182·2 mm. The difference between 194·8 and 182·2, or 12·6, mm. is the amount by which the spring should be shortened.

If instead of a loss we have a gain on the rate, the method of performing the calculation will be identical except that the second and fourth terms will both be larger than in the above example.

If the balance-spring is too short we can, providing the motive force is sufficient, produce a losing rate by increasing the diameter or weight of the balance (**1316** and following articles) by fixing lateral screws on the rim.

1395.—*To determine the length of a balance-spring by calculation.*—If it be required to ascertain the length of a flat spring without being obliged to lengthen it out, count the number of complete turns of which it consists, commencing from the point of attachment to the collet; measure accurately the diameter of the collet and the external coil, and adding these two measurements together and dividing by 2 we obtain the mean diameter; multiply this by 3·1416 and the number of turns and add to the product the length of the fraction that remains of the outer coil, which can always be easily measured. The total thus obtained will, if the measurements and calculations have been made with care, give the length of the balance-spring to a sufficient degree of approximation.

The length of the cylindrical portion of a chronometer spring can be easily ascertained by this method, that is to say by multiplying the diameter by π and by the number of coils; and to the product obtained must be added the length of the terminal curves, which can be easily determined by bending two pieces of waste spring into identical forms and subsequently straightening them.

Or to measure a flat spring it may be placed in a small barrel arranged for the purpose, and the spring coiled on itself round the arbor; but this method does not offer any special advantage.

1396.—*Observations.*—Watchmakers will do well to accustom themselves to these several applications of theory to practice as they will unquestionably very often render important services; but they must not expect to always secure an

exact determination by its means, especially with ordinary watches. In such cases it will be wise never to cut away the waste end of the spring by the full amount indicated by theory, at any rate till after trying it in the watch, because the calculation always assumes the metal to be homogeneous and absolutely uniform throughout its length both as regards transverse section and elasticity; and this is very seldom the case with the rolled springs of commerce, especially the very small ones. At the same time we would recommend practical men to accustom themselves to these calculations; in conjunction with the methods already indicated (**432**, etc.) and others that will be given subsequently (**1399**), they will be enabled without much trouble to plan for themselves a method of procedure that will facilitate their work, enabling them to estimate, very approximately, the gain or loss that corresponds to a definite quantity such as a quarter, half or three-quarters of a complete coil.

To calculate the number of vibrations of a balance as dependent on the strength of the balance-spring.

1397.—We know that the strength of a balance-spring varies inversely with its length, so that if, for example, it be shortened by three quarters, it will exert a force four times as great and will double the velocity of the balance carrying it.

In other words: *the number of vibrations of a balance in a given time is proportional to the square root of the strength of balance-spring.*

So that, in the above example:

The initial strength of balance-spring is . . 1
That of the shortened „ „ „ . . 4
When the first makes . . . . . 1 vibration
The second „ . . . . . 2 „
and 2 is the square root of 4.

PRACTICAL APPLICATIONS.

1398.—In virtue of the above law, if we represent the number of vibrations of a balance by n, and by f the strength of the balance-spring, while N represents another number and F the force corresponding to it, we have:

$$n : N :: \sqrt{f} : \sqrt{F}$$

or $$n^2 : N^2 :: f : F$$

F and f may be replaced by P and p, or the two weights that produce the same angular displacement with the two springs when applied at the ends of equal arms.

Example.—A watch beats 16,200 vibrations per hour and is required to make 18,000 vibrations in the same time. The square roots of the weights that deflect the balance-springs through equal angles will be in proportion to these numbers. Assume the weight in the first case to be 20 cgrm. (3 grains), we have the ratio:

$$16{,}200 : 18{,}000 : : \sqrt{20} : \sqrt{x}$$

Or, observing that 16,200 vibrations per hour is the same as 4·5 in a second, and 18,000 the same as 5, we have, employing the second proportion:

$$4{\cdot}5^2 : 5^2 : : 20 : x$$

$$\text{or } 20{\cdot}25 : 25 : : 20 : x\text{; whence } x = 24{\cdot}6$$

The spring must then be shortened, in other words it must be held at such a point that it maintains equilibrium with a weight of 24·6 cgrm. (3·8 grains) under the same conditions as those under which it previously balanced the weight of 20 cgrm.

1399.—In order to apply these tests a small instrument must be made that will enable us to coil up the balance-spring, the stud being held stationary, and a pivotted arbor, provided with an index and with a thread wound on it to support the weights, passing through the collet.

In practice the force of the spring is ascertained by holding (in the jaws of a small tool made for the purpose) the outer extremity of the spring, while a small weight is suspended to its centre in front of a graduated rule, a method that is precisely similar to the common mode of judging of the strength by the eye, holding the outer end in tweezers while the balance is suspended from the centre (**421**). The cone so formed should consist of a series of coils at regular intervals apart, as if such is found not to be the case it proves that some coils are stronger than others. This method is satisfactory so long as the springs examined are of approximately the same strength but not if the laminæ differ sensibly in section. For then two springs that *apparently* exert the same force will give very unequal rates.

The balance-spring as affected by dilatation.

1400.—We have seen (**1376**) that certain watchmakers attempted to devise a balance-spring that would prevent the escapement being put out of beat through the expansion or contraction of this spring at extreme temperatures. Such an attempt is all but useless for two reasons; in the first place the

spring would be much more difficult of construction and adjustment and, secondly, if the effect were produced, it would be caused by the spring being carelessly made or of bad quality.

On a steel plate A A (fig. 8, plate XIX.) trace out the spiral *s a m d n*. Raise the temperature of this plate gradually and uniformly; it will become larger in every direction in accordance with the law of dilatation (**1259**), and we will assume that its circumference reaches the dotted circle *x x* at some temperature under 100° C.; the enlargement of the spiral unbroken line will produce the spiral represented by the dotted line *i j g f c b*. This line is obviously longer than the unbroken line so that the fixed points of the spring can be transferred from *s* and *n* to *b* and *i*. But, as will be seen, the enlargement of the spring takes place in all directions, and yet, notwithstanding its great increase in length, the displacement of the points *s* and *n* takes place along a straight line and its actual amount is only equal to *s b* and *n i*.

If now we place on the cold plate a very large balance-spring formed of perfectly homogeneous steel, hardened and tempered uniformly throughout its entire length and formed so that it exactly corresponds to the figure traced on the plate, we shall perceive, on raising the temperature in the manner indicated above, that the expansion will take place as here explained, and the coincidence between the spiral line and the actual spring will be maintained throughout.

The dilatation of a hardened spring that has been made with care can only cause the escapement to get out of beat if the expansion of the two arms to which it is attached is excessive; but the difference caused by variations in the radius of a collet can obviously be neglected and, in chronometers, the other arm is made of steel.

1401.—If such an effect is observed to occur, it must be caused by the balance-spring being in a constrained condition through being bound at its points of attachment or by one of the circumstances enumerated in article **1260**. The ordinary springs of commerce, which are rolled and subsequently coiled and tempered insufficiently and unequally, are always in a state of molecular constraint which causes them to open out, often to a considerable extent, on heating. But, as we have already stated, this effect is not observed when the spring has been uniformly hardened and tempered (**351**).

EXPERIMENTAL AND THEORETICAL DATA.

1402.—It seems advisable to enumerate here certain experimental data, confirmed by the theoretical laws, which are due for the most part to P. Le Roy, Ferdinand and Louis Berthoud, U. Jurgensen, Houriet and others, although most of these data have already been given in this work. We will supplement them by observations of our own (*see* also **670—6**).

1403.—The lamina should be of uniform section, homogeneous and of equal hardness throughout its entire length.—The terminal curves should be produced without unnaturally distorting the metal. It would be well, if any fears were entertained on the subject, to bring the metal to a uniform molecular state by a careful tempering. The studs that hold the spring must neither scratch the metal nor constrain it.

A lamina that is non-homogeneous will become distorted on the temperature varying and will not exert the same force throughout its length. The same will be the case if the spring is carelessly pinned in position; under these circumstances, as well as when the curves are in a constrained state owing to their having been too sharply bent with tweezers, the metal will be in a state of permanent internal strain and this will give rise to a progressive gaining rate. The metal will never arrive at a stable molecular state or at any rate not until after a long period, one or two years at least.

1404.—A spring may be found to possess several isochronal lengths corresponding to different balances and to the various periods of vibration, or it may not have a single point throughout its entire length.—Blades will secure isochronism only so long as the extent of the vibration is within certain limits.—If it be required to replace one isochronal spring by another of the same length but different strength, employing a *heavier* balance, the section of the spring must be increased in such a proportion as to secure the required force. Thus, if the width of the blade is doubled, the spring will become twice as strong and will still be isochronal for arcs of similar amplitude.

1405.—If one and the same cylindrical spring (without terminal curves) be re-wound so as to give a greater number of turns, in other words on a cylinder of less diameter, the balance will have greater freedom of movement than before because the angular displacement of the point of attachment to the balance

will take place at a less rapid rate than formerly. If it be materially altered in either direction by coiling the lamina upon a cylinder whose diameter is considerably greater or less, the effect of the spring will no longer be such as to secure isochronism when the arcs of vibration are of the required extent. Thus we see that for a given length of a cylindrical balance-spring, its diameter is not a matter of indifference.

Neither can the points of attachment of such a spring be arbitrarily chosen. With a view to retain its centre of gravity on the axis of the balance, these points are so chosen that the force tending to displace the spring in one direction is exactly counteracted by that which draws it in the opposite direction. This action is complicated (**1436**).

1406.—It is a matter of indifference whether a spring with terminal curves is coiled upon a large or small cylinder; but if this latter is very large, the slightest errors in the formation of these curves will have much more effect on the development of the spring; if, on the other hand, it is very small, it becomes exceedingly difficult to form the curves accurately and at the same time to avoid straining the metal by bending the lamina too much. The two curves may be different.

It is of no importance whether the two terminal curves be opposite to each other or not. They may coincide or be opposite or at any angle. Their form is not dependent on the length or sectional area of the spring, but the length must be sufficiently great to prevent all risk of distortion within the range of the longest arc of vibration.

1407.—In conclusion, a spring that is in any way damaged, by unequal hardening, bends, twists, parts that have been strained and bent back into position, inferior metal, a constrained state due to the points of attachment or carelessly made terminal curves, etc., will rarely secure isochronism in the first instance, and, even if it does, it can never be relied upon in the future.

We would repeat that the hardening and tempering of an isochronal spring require very great care. If the hardening has been uniformly effected, the timing will not be rendered more difficult by tempering too much or too little. When a spring is too much tempered the formation of the terminal curves is more easily effected but there is also a greater risk of its being distorted by vibrations that are of too great extent. This danger is avoided by tempering the spring hard. It is

true that in such a case a more marked gaining rate is observable in the chronometer, but this progression gradually becomes less and will altogether vanish in a short time, as already observed.

The importance of thus having a spring in a *stable* condition, that is to say with a molecular arrangement that is perfectly uniform and not subjected to any constraint, was observed in the early days of chronometer-making by P. Le Roy (**1382**).

He added: The spring must not have too great length or mass in proportion to its strength, for then any shake gives it a vibratory movement, not only throughout its length but also transversely to the lamina.

1408.—After a certain number of years' service chronometer balance-springs are usually found to be useless; their elasticity has altered, very often without its being possible to detect any apparent cause. It is generally assumed that such deterioration is due to a kind of fatigue of matter.

Traces of rust deprive a spring of all its valuable properties.

To re-form the terminal curves.

1409.—The article on Timing explains the mode of selecting an isochronal spring: we will only consider it here as fixed in the chronometer, which is going.

As a general rule and in ordinary cases, if the chronometer loses in the long arcs, in consequence of the force exerted by the balance-spring not increasing in a proportion that corresponds with the angular displacement of the balance, the spring is shortened and the terminal curve re-formed. If only a very slight correction is needed it may be effected by modifying the form of curve; most frequently in such a case it is opened out or rather rounded off, and it is made to turn up more rapidly from the point of attachment, so as to cause the spring more or less to oppose the ascending tendency of the balance.

When the chronometer gains in the long arcs, the spring is lengthened and the terminal curve re-formed, if a marked change is requisite; but if only a slight alteration is necessary merely the terminal curve itself is re-formed, its curvature being modified so that it bends more insensibly towards the point of attachment and closer in to the centre; in other words so as to facilitate the motion of the balance towards the conclusion of its period of ascent.

If these data have been clearly understood the intelligence of the reader will supply the rest, since a book cannot replace

practical experience. Let the watchmaker study with care the theoretical conditions of isochronism and the causes that modify them in practice, and, after mastering the numerous problems treated of in this work, let him make a number of observations. He will thus be enabled to realize, doubtless not without some difficulty at first but with no less certainty, the conditions of practical isochronism (**1388**).

The flat balance-spring.—Limits within which it is isochronal.

1410.—The flat balance-springs of ordinary watches are comparatively short so that they coil up at a too rapid rate; this causes a watch in the great majority of cases to gain when the oil is fresh, because the vibrations are then more extended in consequence of the temporary increase in the motive force. It will no longer be the case when the oil becomes thick, the spring enervated, etc. Thus it is only possible in such cases to secure a sort of chance isochronism, due to the fact that there exists a certain definite proportion between the length of balance-spring, extent of vibration, friction of the pivots when forced laterally against the sides of their holes, and the various effects due to thickening of oil in the escapement.

Within the limits between which it is possible to secure isochronism with a flat spring it can only be realized if the following conditions, deduced from theory, are observed.

The internal point of attachment should be as near as possible to the centre of the spiral spring and from this point the lamina should graduate from the collet insensibly and with very great regularity. For otherwise the progressive movement of the spring will not be such as to secure isochronism.

Suppose for example that a portion $a\,a'$ of a balance-spring (fig. 10, plate XXI.), instead of having only a single point in common with the circular arc described from the centre of the spring, coincides with the portion $b\,b'$ of this circular arc. It follows that when the balance impels the portion $a\,a'$ in the direction from a' towards a, it will meet with less resistance than in returning; for in that case the portion $b\,b'$ will, so to speak, butt directly against the action of the balance; whence it follows that between the advancing half of the vibration and the return vibration, irregularities will be met with that are incompatible with the conditions imposed by the law of isochronal oscillations (see also the *Watchmaker's Handbook*).

1411.—The flat balance-spring (when it is sufficiently long and isochronous) is exceedingly convenient for use in the best class of watches, and when the arc of vibration is not of too great extent it is often preferable to the Breguet spring. There is greater difficulty in forming this latter so as to conform to the conditions required for isochronism, as the steel is strained and distorted at the bend, besides which this bend interferes with the uniform progression of the flexure, which is there complicated by a tendency to twist.

It would be well, in the case of ordinary lever watches, to replace the great majority of the bad compensation balances, whose only characteristic consists in disturbing the equipoise of the balance with every change of temperature, by ordinary balances fitted with screws on the circumference; for such balances can be much better brought into accord with the motive force and with a balance-spring that is sufficiently isochronal for ordinary purposes.

Movable Stud.—Its effect in increasing the range of isochronism with the flat spring.

1412.—The principal objection to the ordinary balance-spring arises from the fact that the centre of the cock is far removed from one point of attachment of the spring, for, in consequence of this, the main body of the spring is compelled to be thrown suddenly to one side when the vibrations are even only of average extent. This gives rise to a temporary distortion which interferes with the true isochronal development.

This fault can be avoided by attaching the outer extremity of the spring not to a stud fixed in the cock but to the end of a straight spring that is fixed by a foot to the side of the cock or to the plate of the watch as shown in fig. 18 (plate XIX.). Such an arrangement is known as an *elastic* or *movable stud.* At each vibration of the balance this stud is deflected so that when the balance-spring closes its outer end is drawn nearer to the centre and this again moves outwards when the spring opens. The arrangement, which is due to Young and Hardy, helps to extend the range of the isochronal development of the flat balance-spring.

The principal difficulty consists in determining the exact dimensions of the elastic stud: for it is evident that, with a given spring, it must satisfy certain conditions as regards length, thickness, elasticity, etc., which as yet can only be ascertained by experiment. Moreover, strictly speaking, it is

essential that the head of the stud be almost without weight, in order that its elasticity alone may act and that its weight may not have different effects in the vertical and horizontal positions.

This subject is however still too novel for us to further consider it; we would only add that C. Frodsham employed a flat balance-spring and elastic stud in a marine chronometer which ultimately proved to be one of the best he produced. M. Raby of Paris has also employed them in his watches and he found them to give satisfactory rates.

TO MAKE AND HARDEN A CYLINDRICAL SPRING.

1413.—The old watchmakers made their balance-springs of round steel wire, previously flatting it by passing between a small pair of rolls. Even at the present day wire prepared for this purpose is to be met with, which only needs to be coiled into the requisite shape and hardened.

The diameter of the spring being known, either by measurements taken from a chronometer that serves as a pattern or by preliminary trials, a hollow cylinder or block is prepared on which the wire must be wound in the required cylindrical form in such a manner that it may retain this form after hardening (the terminal curves are made afterwards with tweezers). Attempts have been made to harden the spring on the block after having formed these terminal curves, but they have rarely been successful because the spring becomes longer, the curves open out more or less and it would be necessary not only to form curves on the block different from those ultimately obtained, but the expansion of this block would have to be absolutely identical with that of the spring, so that it would be very difficult of construction.

1414.—*To make the block.*—This is generally formed of copper or brass.

The block must not be passed through a screw-plate and the wire afterwards coiled in the thread; such a method would give the very worst results. Unless the tube was much too thick it would be distorted in the tapping; and moreover the bottom of the hollow does not lie throughout on the same cylindrical surface, and the metal which is subjected to such rough treatment as the cutting of a screw will be certain to suffer distortion when heated, etc.

The block is generally hollow; some trace a thread upon

it in the machine for cutting fusees, others leave the external surface perfectly smooth.

It should be accurately cylindrical both internally and externally. If a thread is cut upon it this should be done with a very clean cutting cutter that removes only a small quantity of metal at a time and polishes the bottom of the groove, which must be shallow. When this is the case the metal will have a uniform thickness throughout, and, if it be gently heated, it will, when surrounded by the balance-spring, receive the heat uniformly without undergoing any irregular expansion.

The metallic shell should be only of a moderate thickness, about 1·5 mm. (0·059 inch). If it is too thick the balance-spring will become red before the block and will cool more rapidly on its external surface; and, as a consequence, the internal and external faces of the spring will not be hardened to the same degree.

When employing the smooth block some makers coil the wire so that all the turns are in contact and fasten the two extremities with screws. The expansion of the spring in the hardening, which may be facilitated by slightly separating the several coils when the band is stretched on the block prior to tempering, will suffice to maintain the turns at such a distance apart that they shall not touch during the vibrations of the balance. Other makers coil two wires side by side, one of which merely serves as a guide to retain the coils of the actual spring equidistant.

The block is fastened firmly to the axis of a mainspring winder and one end of the wire is fixed under the head of one screw; a sufficiently heavy weight is then attached to the free end in order to stretch the wire, and the coiling is effected by rotating the block until the wire reaches the second screw when it also is fixed, and the whole is then ready for hardening.

1415.—*Mould for making spherical springs.*—The mould employed by F. Houriet in making his spherical balance-springs consisted of an axis formed of two parts fitting one into the other, on which were adjusted five brass discs held together by lateral screws. He turned this system into a sphere flattened at the poles and traced a thread on it with the fusee engine; tracing first one half and the other after reversing the axis. After coiling and hardening the balance-spring on this mould he removed the screws, divided the axis and removed the brass discs one at a time.

Motel formed the core of twelve discs accurately fitted on

a square axis and held together by a screw; on removing the screw and axis these discs could be removed through the intervals between the coils.

1416.—*To harden a cylindrical balance-spring.*—The main element of success consists in a knowledge of the degree of heat best suited to the steel employed, and in the skill of the workman in applying the heat and in seizing the precise instant at which the object should be immersed in the liquid. It requires considerable practice in hardening to secure springs that are elastic, equally hard throughout their entire length, and that remain of a uniform white colour after the process.

The usual practice is to suspend the block by an iron wire that traverses it and then to bed the block in animal charcoal in a tube or crucible which is placed on the fire. If it is possible to rotate the crucible so that the heating may be more uniform, all the better. When it has been raised to the requisite temperature, the crucible or tube is rapidly removed and placed by the side of a vessel containing oil and, by means of the iron wire, the block is quickly removed and dropped into the oil. If the vessel is not of considerable depth, the iron wire should be held in the hand and the block moved from side to side in the liquid. Some makers merely empty the crucible into the liquid contained in a vessel from six to eight inches deep.

Observation.—The animal black should be free from impurities and thoroughly dried, and the spring perfectly clean and dry; neither should be touched by the fingers, for particles of the black would then adhere and occasion stains.

1417.—*F. Houriet's method of hardening.*—This has been already described in article **497**, page 285.

Motel's method.—Motel enclosed the spring, coiled on a platinum block, in an iron tube which was then filled with a mixture of 1 part of powdered burnt leather and 3 parts animal black, and he hardened in water. The springs prepared in this manner were extremely hard; they were tempered on a block of copper slightly larger than the first.

M. Rozé's method.—He detaches the spring from the block before hardening and places it in a bulb of green glass which, after being filled with hydrogen, is hermetically sealed in a blowpipe jet. When heated to the requisite temperature the whole is thrown into water, the bulb breaks and the spring is exposed without having been in contact with the air.

1418.—*To clean and temper a cylindrical balance-spring.*—The spring is carefully removed from the block, on which it is loose, having been lengthened by the hardening; the metal is then polished on the inside by rolling it gently with a truly cylindrical roller of hard wood, very nearly in the same manner as the inside of a cylinder is polished. Very great care is necessary in consequence of the brittleness of hard steel.

The spring is next put on a smooth block. After fixing one end under the first screw, the block is rotated, at the same time gently pressing on the lamina so as to tighten it a little, and the other end is then fixed with the second screw. The sides of the spring are cleaned by holding a piece of wood between two coils and rotating the block. And the external face is similarly cleaned by laying the wood flat.

If the hardening has been properly conducted the spring should be left perfectly white, so that it is not necessary to remove any discoloration from the surface: dry rouge on wood will generally suffice in such cases, but otherwise very fine emery, etc., must be employed.

When an iron or zinc polisher is used for the outer face of the lamina, there is a danger of removing more metal from some points than others, so that the progressive force, necessary for isochronism, is destroyed.

When the entire surface is perfectly clean the balance-spring may be tempered: some stand the block upright on the blueing tray inverting it at intervals. Others support it on a rod that is caused to rotate while one end is in a flame; and others again roll the block on a very smooth plate previously heated to the requisite degree.

These methods are not satisfactory; it is preferable to place the block in a metallic box which is set on a charcoal or turf fire that is not too hot but very uniform and protected from currents of air. Put a cover on the box and a piece of white steel on this, as a means of observing the progress of the operation. Or the box might be suspended above the fire and caused to rotate as in hardening mainsprings.

One maker, after hardening the balance-spring, tempers it in oil which is heated to a temperature that experience indicates to be necessary; on its removal the spring has a bronze colour which can be removed in the manner above indicated if required, and the spring can be subsequently blued.

After being cleaned in pure alcohol and dried it is ready for use.

To make and harden a flat balance-spring.

1419.—Formerly the flat balance-spring was made entirely by hand. Holding the steel wire loosely between the thumb and index finger of the left hand so that a small portion projects, the end of this is taken between the thumb of the right hand and a broach, the handle of which is grasped by the four fingers of this hand. With the two hands pressed together, the right hand, partially inverted, draws towards itself with a circular movement small portions of the wire, which is gradually coiled up by the semicircular movement of the right hand and the pressure of the thumb keeping the wire in contact with the broach.

This work was generally performed by women who coiled the springs with extreme rapidity. After the wire was bent into a spiral form they placed it alternately on either face on a card, retaining it with the fingers of one hand and raising coils that were bent downwards by means of tweezers held in the other hand.

1420.—The above method is very seldom employed at the present day, being replaced by the following. A brass plate of the form shown in fig. 12, plate XIX., is perforated at the centre and hollowed out as though it were a small barrel, except that the rim is cut through at several points. Another plate, which is not hollowed but usually cut away like the crossings of a wheel, is employed to cover the first-named plate, being held in position by the screw *c*. The two plates when together fit on to an arbor *m*, which is held by friction and can be removed when required. The extremity *n* of this arbor is traversed by one or more grooves. One or more lengths of balance-spring wire are passed through the openings in the rim and the slits in the arbor-head, which is then rotated until the coiled up wires entirely fill the space between the two plates; the projecting ends of the wires having been cut off and the arbor removed, the brass envelope containing the springs is placed on the blueing tray and heated until a blue colour would be imparted to steel. One or more springs can be made by this method according to the number of wires employed, and the distance between the coils will be determined by the number of balance-springs so made.

The observation just made with regard to the distance between the coils is only true when they have been hardened or at any rate very highly tempered together; for the bobbin wire

is merely drawn through the draw-plate, in other words hardened by a rolling action and the spring always opens out more or less on being removed from the barrel, all the more according as it has remained a less time in it or has been less highly tempered. It is a frequent practice only to coil one wire in the barrel on itself, and, on removing it, the coils will separate of their own accord. Drawn steel usually loses a little of its elasticity when tempered after coiling; but its molecular state will be made more stable by the process.

1421.—*To harden a flat balance-spring.*—Although the above explanations seem to be sufficient to enable a watchmaker to harden a flat balance-spring when he finds himself under the necessity of doing so, we will give a few further explanations.

He may either harden a number at once contained in a small barrel as already stated, or they may be placed between two plain metallic discs that are held together by a screw at the centres. The whole is covered with wood charcoal or fine animal black that has been carefully dried. Or a single spring might be placed at the bottom of a shallow tube and animal black sprinkled over it in such a manner as not to displace the coils. After hardening it will be only necessary to temper the spring between two plates if any coils are out of flat, and a balance-spring of good colour will thus be obtained.

Frederick Houriet constructed a machine by means of which he cut a spiral groove in a platinum plate. In this he set the wire, subsequently hardening and tempering it; but, as will be evident, the method is wanting in facility of execution.

If the spring is not of a good colour after hardening it must be tempered in oil, afterwards passed rapidly through dilute sulphuric acid in order to whiten it and again blued. But this method must be practised with caution after several trials and in no case must the action of the acid be too rapid.

Balance-springs of alloyed gold.

1422.—Breguet,* Houriet, and more especially U. Jurgensen and his eldest son, employed gold balance-springs in

* Breguet was born in Switzerland in 1747 of refugee French parents and died in Paris in 1823. Endowed with considerable ingenuity and a taste for complicated and remarkable mechanisms, with ready wits and high patronage, this artist, in whose workshops the best workmen of Europe were employed, enjoyed during his lifetime more fame as a horologist than any other maker of the present century. He was a member of the Institute of France although less learned than horologists that were either his contemporaries or predecessors, such as P. Le Roy, F. Berthoud and A. Janvier.

their chronometers. Jurgensen states that when they have been well hammered they will, after very considerable tension has been applied, accurately resume their original form and elasticity. He finds them advantageous in that they avoid all the effects of oxydation and magnetization. It is undoubtedly true that gold expands more than steel and therefore requires a more marked compensation, and that it is heavier and would on that account be unadapted for use in pocket chronometers, owing to the fact that the spring would tremble more easily.

Alloyed gold that has been made hard by hammering or rolling will become harder on tempering. It assumes the same tints as steel successively with the exception of blue which is replaced by a grass-green colour.

Some temper it by immersing in boiling oil for a length of time that is determined by experience; others place the block carrying the spring between two burning pieces of turf at the part where they are reduced to white ashes at the surface; the introduction of the block breaks them, so that it is immediately covered with the ash which must not be removed until quite cold.

Springs that have been hardened in this manner will break just as steel springs and the hardness is all the greater as the proportion of gold is less.

Another mode consists in enclosing the block carrying the spring in a tube, on the top of which is placed a piece of polished steel. The whole is heated in a spirit lamp and, when the steel assumes a blue colour, the lamp is extinguished.

A question has frequently been raised as to whether gold springs are preferable without any solution being arrived at. We would ask the two following questions of those who may in future occupy themselves with this subject:—Will the coefficients of elasticity of two laminæ, one of alloyed gold and the other of steel, change in the course of time in different proportions? —A steel lamina that is always in a state of vibration changes its molecular condition either with the lapse of time, or through the continued action of causes that are known or are still unknown; is the same the case with a lamina of a gold alloy?

SPRINGING AND TIMING.

ACCURATE ADJUSTMENT.

1423.—The compensation balance generally employed in marine chronometers is shown at fig. 7, plate XXI. A A are

the compensation weights, B B, the timing screws. The other screws, *v v*, are supplementary to the timing screws and facilitate the poising of the balance.

Fig. 8 represents the ordinary compensation balance with screws, such as is employed in watches. There are twenty or thirty holes drilled in the circumference, which render it possible to set the screws in such positions as to give the requisite degree of compensating effect.

These holes are usually set nearer together as they approach the extremities of the arms so that when it is required to increase the compensating effect the greater number of the screws can be collected near these extremities. The two screws at the ends of the diametral arm are timing screws.

For accurate timing a watchmaker that does not reside near an observatory where the exact time can be obtained, must know how to ascertain it by observation with the transit instrument; or, when only adjusting ordinary watches, by a meridian dial.

He must be provided with:

A good seconds regulator whose rate should be verified about every eight days.

An oven for testing at elevated temperatures, and an ice-box for experimenting in the cold.

Prepared sheets on which to record the rates.

As well as several tools and instruments:—for fixing the watches in varying positions;—for poising balances on their pivots;—for trueing them;—for making small metallic washers to adjust under the screws so as to increase their weight, etc. He must also be provided with strong pliers with wooden handles to be used for heating the terminal curves and shaping them when so heated. The curved faces of the pliers must not have any sharp angles and their curvature should be rather more marked than that of the spring, since the latter opens out slightly on being released.

Selection of a balance-spring.

1424.—After choosing a spring of the requisite strength, regarding the number of vibrations per hour, and of such a thickness that its sensibility to variations of temperature approximates as nearly as possible to that of the balance, the older horologists employed the "elastic balance" to study the progress of its force (**1379**). This method is satisfactory and enables us

to select, with but little delay, a spring that approximately satisfies the required conditions; but at the present day, when no one does anything but copy or modify springs that experience has proved to be satisfactory, it is preferable to select a spring in accordance with the conditions indicated above and of the same form as one that has been tested, and to try it at once in the chronometer itself (fig. 14, plate XIX., represents an ordinary chronometer balance-spring).

The spring having been selected is attached to the balance. If a novel arrangement of chronometer is under examination, the balance must be made light rather than heavy, in order to test the facility afforded for effecting the final adjustment by varying the total weight and moving the timing screws.

A chronometer should be rated to what is very nearly mean time; by preference there should be a slight gain if the balance is at all light, and it is a good precaution to submit it several times in succession to temperatures about 0° C and 30° C in order the more expeditiously to bring all the pieces to a uniform molecular condition and to cause them to act regularly.

1425.—When testing in heat and cold the temperature should be made to alter gradually; sudden variations cause a deposition of moisture on the metallic surfaces. Moreover sudden dilatation will have the effect of weakening the metal at certain points, it will no longer be precisely in its initial condition and at times will not return to its original position.

To test the isochronism.

1426.—The isochronism should always be tested at a constant mean temperature of about 15° C because the balance has not yet been adjusted to compensate and variations of temperature might therefore occasion changes in the moment of inertia of the balance from which erroneous conclusions would be drawn. The arc of vibration is made to vary from 180° to about 540°.

1427.—First consider the chronometer with going barrel.

Allow it to go in a horizontal position, with the mainspring fully wound up, so that the arc of vibration is a maximum, for 12 or 24 hours. Observe its rate very closely, comparing it with a regulator that is accurately timed.

The mainspring is now let down so as to make the amplitude of the vibration less than it would be under ordinary circumstances after a given lapse of time, three or four years or

even more, and the chronometer is made to go in the same position for a similar number of hours.

If the rate is absolutely the same with the short arcs as it had previously been with the long arcs two extreme isochronal points have been found.

But if not the length of spring must be modified in the manner indicated by the rates (**1409**), and the tests repeated. It will of course be understood that if the length of the balance-spring be modified it will give rise to a gain or loss in the twenty-four hours; but with this gain or loss we are not concerned unless it becomes excessive, as it can be counteracted subsequently by means of the timing screws (**1431**).

If the spring has no terminal curves, not only its length but also its diameter is modified (**1405**).

If there are terminal curves it will only be requisite, when a slight modification is needed, to modify one of them.

1428.—When a balance-spring has been selected approximately corresponding to the required number of vibrations in an hour and when two extreme isochronal points have been determined, the isochronism of the intermediate arcs must be tested, gradually increasing them by small fractions: for, since the lamina is rarely quite homogeneous, equally hardened throughout, etc., if the vibrations successively tested differ by too great amounts from each other, errors may exist in an interval between two amplitudes that were duly examined. It must be remembered that there is occasionally a difference observable between the progressive force of a spring when being wound up and when running down.

1429.—Another method is resorted to when experimenting with a fusee chronometer, which is however equally applicable to the going barrel; this mode consists in coiling a fine cord round the barrel drum and suspending a weight to its extremity; or the following may be adopted:

Fix the chronometer in a horizontal position after having fitted a pulley on to the winding arbor; on this pulley wind several coils of a fine cord which should be made to pass over a guide pulley, a weight being attached to its free extremity. The weights used are so graduated as to produce different amplitudes of vibration of the balance.

The weight of the balance depending, as we know, on the motive force, it will be necessary in the case of a chronometer

of novel calliper to experimentally ascertain the ratio between this weight and force, having regard to the conditions of the lift; in other words experiment on variations of the three terms, motive force, weight of balance, and conditions of the lift.

1430.—*Observation.*—When the several conditions on which the action of the balance-spring depends are suitably co-ordinated we shall have one that is perfectly isochronal; but experience has demonstrated that a slight gaining rate in the short arcs is beneficial, since it helps to counteract the loss that time brings about in the rate of a chronometer when the arcs are shortened; so that, instead of being absolutely isochronal, a balance-spring is needed that will occasion a slight gain in the short arcs; and the amount of this gain should gradually diminish as we increase the amplitude of the vibrations.

It is impossible to fix *a priori* the amount of this acceleration in the short arcs; it depends on the form of escapement adopted. Earnshaw and Jurgensen made it six seconds in 24 hours when arcs of 450° were reduced to 300°. L. Berthoud and Motel never exceeded about three seconds as a mean.

We have examined chronometers that had maintained excellent rates at sea where the acceleration was as much as eight or nine seconds and others where it was nothing.

As a general rule too much care cannot be devoted to the construction, setting in position and management of an isochronal spring, and it is only by a careful study of the properties of curves and of the changes introduced in the rate by different modifications in their form, that the watchmaker can expect to be thoroughly versed in his subject and can master any given type of instrument (see articles **670—6**).

Adjusting by the Timing Screws.

1431.—In testing the isochronism it was only necessary that the chronometer went approximately to mean time. After the test is completed, the watch is made to keep exactly mean time by adjusting the timing screws: screwing them in so as to bring them nearer the centre of the balance if the watch loses, and the converse if it gains. The total weight of the balance is of course maintained unaltered by this adjustment.

If needful this weight can be increased or diminished by adding or removing screws near to the timing screws; that is to say in that part of the rim that remains practically unaffected

by variations in the temperature. A moderate change in the weight will have no appreciable effect on the isochronism.

To adjust the compensation.

1432.—The bi-metallic arcs must be so far sensitive to variations in the temperature that they are not behind the balance-spring in undergoing change. They should be broad and of moderate thickness rather than narrow and thick. The precise thickness must depend on the amount of elastic reaction that is to be anticipated as well as on the influence of centrifrugal force in tending to open out the arms in the long arcs.

Balances usually have two bi-metallic arms but they are also occasionally made with three and four arms. The latter have the advantage of occasioning a less amount of loss at extreme temperatures; but, as M. Rodanet observed, they necessitate the use of a brass containing more zinc since otherwise the compensating effect is insufficient. They are also more difficult of construction.

After carefully timing the chronometer at some mean temperature (say 15° C or 60° F) it is generally subjected, for 12 or 24 hours each, to the extreme temperatures 0° C (32° F) and 30° C (86° F), or only to 5° C (41° F) and 30° C.

If the instrument gain in the heat and lose in the cold it is over-compensated; the weights must then be moved backwards, that is to say towards the fixed ends of the bi-metallic strips. For the weight will then be in such a position that it is moved through a less distance, and the amount of compensating effect will be proportionately less than that previously observed.

An opposite movement will be required in the converse case.

1433.—This method is sometimes found to be insufficient and it may become necessary to replace the weights by others that are heavier or to diminish them materially; this will involve a corresponding inverse change in the weight of the timing screws. If owing to these several alterations it becomes necessary to alter the length of balance-spring or change it in any other way, the timing must be done entirely over again.

The adjustment of the compensation in a chronometer that is simply copied from one that has already been experimented on is a matter of great delicacy; it may be judged then how difficult it is to devise an entirely new calliper.

If its compensation is not perfect a chronometer becomes useless for the purpose of scientific observations.

Timing in the four Positions.

1434.—The object of this proceeding is to verify the poising of the balance and to ascertain how far a change of position modifies the isochronism and compensation.

The balance cannot possibly be accurately poised in all positions if the pivots and pivot-holes are not perfectly round, and the poising will be modified with a change of temperature if the two arms do not act identically; as will be the case when the metals are not homogeneous, when one or both arms have been strained owing to want of skill on the part of the workman or careless work, etc.

After accurately timing the chronometer in a vertical position with XII uppermost, a position which may be indicated thus $\frac{\text{XII}}{\text{VI}}$, make it go for 12 hours in the position $\frac{\text{VI}}{\text{XII}}$ and the same number in the positions $\frac{\text{III}}{\text{IX}}$ and $\frac{\text{IX}}{\text{III}}$. Observe with care both the rates and the amplitude of the arcs and note them down.

Assuming the pivots and pivot-holes to be perfectly round and in good condition and that the poising of the balance has been previously tested with care by the ordinary means, if the variations in the four positions are slight the poising may be regarded as satisfactory.

1435.—As a general, but not invariable, rule, a loss in one position on the rate observed in the inverse position may be taken to indicate that the weight of the upper part of the balance is excessive when it does not vibrate through an arc of 360° or the lower part if the amplitude exceeds this amount. Independently of the balance this loss may be occasioned by: excessive friction of the pivots due to a too great pressure owing to the calliper being faulty, or to a distortion of the balance-spring causing its centre of gravity to lie out of the axis of the balance. If these influences become at all considerable their correction will be beyond the power of the isochronal balance-spring and indeed it will be impossible to counteract them. The very many designers of callipers who have neglected to study the laws of mechanics would do well to take warning from these facts.

1436.—With a given strength of mainspring differences in the arcs in the four positions indicate moreover that the resistances differ, but the cause of this fact is not always easy

to discover. Besides differences in the distribution of the weight and in the position of the centre of gravity of the balance-spring, which may be either above, below, or on one side of the axis, it must be remembered that in one position the escape-wheel *lifts* the balance and so reduces the pressure of its pivots, whereas on inverting the instrument the wheel increases this pressure; and, in addition to this, the action of the balance-spring will be more or less opposed to the action of the wheel at this instant and will increase the effect, etc. The position of the points of attachment of the balance-spring without terminal curves must then not be left to chance in the case of portable timekeepers, nor even when terminal curves exist that are not rigorously in accord with theory (**1405**).

1437.—It is only by an intelligent study of the subject that a judicious combination of these various elements or effects can be arrived at; that is to say, such a combination that the variations in positions shall not exceed the amount that the isochronism of the balance-spring may be expected to counteract; for in getting some advantage from it we must not exact too much.

The verification in the four positions should first be made at a constant mean temperature for the purpose of testing the poise of the balance and of studying the friction, pressure, etc., by observations on the amplitudes of the arcs. Subsequently they may be repeated at the extreme temperatures in order to ascertain that the poise of the balance is maintained and to see how far the friction, etc., are modified by a change of temperature.

Timing in vertical and horizontal positions.

1438.—Our object is to give some useful indications as to the best forms of pivots, of the sides of the pivot-holes, etc., for maintaining the rate of a watch unaltered when moved from a vertical to a horizontal position or *vice versâ*.

The intensity of friction is proportional to pressure and independent of the amount of acting surface (**37** to **41**), and therefore it can only vary with whatever occasions the pressure; but it is important to take account of the resistances due to differences in the radii of friction, to the adhesion and to the thickening of oil, which are proportional, the first to the lengths of the radii and the others to the extent of surfaces in contact.

If we consider the two cases of the axis being horizontal and vertical, we see that: in the first case the friction at the two pivots takes place between two curved surfaces of the same

radii as the pivot, and, in the second, the friction occurs on the surface of a disc more or less flat whose greatest radius is equal to that of the pivot and it gradually diminishes from the circumference towards the centre. On investigating the nature of the friction in the two positions we observe that: (1) the frictional surface is almost always of greater extent when in the horizontal position; (2) in this position, even if the acting surface is the same as when vertical, the resistance will be greater; for the friction all occurs at the extremity of a long arm whereas in the other position the radius of friction diminishes up to a point at which it is zero; (3) the resistance due to adhesion between the oiled surfaces is always greater in the horizontal position.

1439.—Assuming there to be a progressive increase in the weight of the balance, we see that, if when the axis is vertical the resistances increase in proportion to the weight, they will increase in a more rapid ratio with the axis horizontal, and we can thus explain the fact, to which Jurgensen drew attention, that when the balance is too heavy the watch is difficult to time in the horizontal and vertical positions, as well as the following observation of F. Berthoud:* "If the motive force of a watch is too great in comparison with the force exerted by the balance, the watch will gain in the horizontal position and lose when placed vertical." It must be borne in mind that, as we have already demonstrated in articles **647—651** and in **666**, the force exerted by a balance varies in a different proportion from the motive force (**1352**).

The weight and motive force, then, cannot be fixed upon arbitrarily. By making them vary a combination will be arrived

* Ferdinand Berthoud was a Swiss horologist, born in 1727 and dying in 1807, and he came to Paris at the age of 19, never afterwards leaving it. His technical training was matured and perfected by contact with the great masters of that day, of whom he subsequently became a rival. So far from having *invented chronometers*, as several of his biographers have asserted, he had no success whatever in the art of the chronometer-maker until he adopted the recommendations of P. Le Roy, after having been so dishonest as to endeavour to appropriate several of the most important discoveries of that famous horologist. Although wanting the intuitiveness of true genius, F. Berthoud was possessed of a very extensive knowledge, and real talent coupled with indefatigable energy; these are sufficient to explain and justify his great reputation. He published ten quarto volumes in which unfortunately valuable work is too often ruined by contradictions and assumptions, such as a modern writer has with some reason termed "musings in mechanics." Although these criticisms are abundantly justified, Berthoud did much valuable work and his name will therefore long remain one of the glories of the horological art.

at which gives a very slight difference of rate in the two positions although the motive force vary considerably (of course assuming the balance-spring to be well adapted to the balance).

In the above discussion of the subject we have ignored the very slight amount of friction at the circumference of the pivots when the axis is vertical and of their extremities when it is horizontal, as well as the effects of capillarity, which are unimportant except in the case of small light balances.

1440.—The resistances above referred to are equalized by chamfering the pivot-holes and making the ends of pivots perfectly flat; but it should not be forgotten that, when the acting surfaces are too small for the pressure they have to support, they force the oil away and show signs of wear in a comparatively short time.

1441.—Changes in the rate on changing from the vertical to the horizontal position may also arise from the following causes which have been already in part enumerated in articles **1434—7** on timing in the four positions, and several of the paragraphs are applicable in the present case:

(1) The action of the escape-wheel, which is different according as it tends to raise the balance-staff or to force it laterally; (2) a balance-spring that starts to one side and so displaces its centre of gravity; a balance that is not well poised; pivots or pivot-holes that are not perfectly round, faults which although of but little importance in the vertical position of the balance-staff become serious when it is horizontal; (3) the more marked portion of the friction of the pivots may take place against substances of different degrees of hardness in the two cases, the end-stones being frequently harder than the jewels.

In this case then the conclusion is the same as in article **1437**, and we would add that, when necessary, we can in part employ the method of article **431**. (See The Watchmaker's Handbook.)

CONCLUDING OBSERVATIONS ON TIMING.

Timing in inclined and vertical positions.

1442.—A defective poising of the balance, which is insensible or nearly so when the staff is vertical, becomes more and more apparent as it is inclined towards the horizontal position, where the variations due to bad poising attain a maximum.

M. Phillips has demonstrated, in a Memoir presented to the Academy of Sciences, that the amplitude of vibration of the

balance that reduces the variation in the rate due to imperfect poising in the various inclined positions to a minimum is 439° 28′, or, in round numbers, 440°; rather less than one turn and a quarter. The following table gives the results of his experiments:

Number of Apparatus.	Number of Experiment.	Amplitude of the vibrations of the balance.	Maximum variation in the daily rate according to the inclination of the steff.
I.	1	270°	284·0 seconds.
	2	440°	3·0 „
	3	480°	32·2 „
II.	1	270°	361·0 „
	2	440°	12·0 „
	3	455°	73·0 „
III.	1	260°	532·8 „
	2	440°	24·0 „
	3	470°	165·0 „
	4	360°	185·0 „
IV.	1	360°	138·4 „
	2	440°	6·3 „
	3	475°	57·3 „
V.	1	195°	122·7 „
	2	440°	3·3 „
	3	530°	122·4 „

It will be well, in the case of pocket chronometers, to approximate as near to this number of degrees as the other conditions that have to be satisfied will permit; but the important conclusion to be drawn from M. Phillips' demonstration is that it proves to us that, since the arc of 440° is the least sensitive to defective poising, we must carefully observe the rate in various positions when the amplitudes differ materially from this amount; for otherwise the maker might conclude that the poising of a balance is perfect when in reality it is very deficient in this respect.

Deviation from perfect isochronism after a time.

1443.—Metals that have been worked, that are constantly vibrating or are subject to friction, generally require a certain time to elapse before they arrive at a condition that is practically permanent. A new watch that is accurately timed in the first

instance is rarely so a year afterwards; a fact that may be easily verified by testing the rate.

As a general rule in new instruments the isochronism of the balance-spring will, even after the lapse of only a few months, be partly neutralized by changes in condition either of the balance-spring itself or of other parts. During the first year this spring is frequently under the influence of such forces as that of crystallization (at least this is the opinion of very many authorities), and they give rise to a progressive gaining rate of the chronometer; this gradual increase ceases after a time (**1374**).

On accelerating the short arcs.

1444.—The older chronometer-makers ascertained that a considerable number of marine chronometers, provided with perfectly isochronal balance-springs, were characterized after a time by a slightly losing rate; that is to say when the arc of oscillation was somewhat reduced. They attributed this retardation to a falling off in the elastic force of the balance-spring and to the thickening of oil; two causes which in their opinion would be competent to make the rate of movement slower in the short arcs notwithstanding the isochronal spring.

An increase in the resistance at the pivots has, in other words the effect of counteracting or rather masking the isochronism: this can be demonstrated in most chronometers that are provided with perfectly isochronal springs by observing their rate in the vertical and horizontal positions before they have been adjusted for positions; they will lose in the short arcs when vertical, and the retardation will be the more marked as the temperature is lower.

1445.—The explanation given above of the observed facts does not seem to us satisfactory for several reasons.

(1) Marine chronometers do not vary their position, they have heavy balances, fine pivots, and, when the holes are well made, the oil does not offer any appreciable resistance to the pivots, for it remains still fluid at the end of two or three years (but it will be easily seen that such is not the case in the train on which depends the energy of the impulse).

(2) The same cause must always produce similar effects; but how can we explain an observation that has been made on numerous chronometers which have maintained perfect rates when at sea? In some the isochronism of the balance-spring was absolute whatever the arc of oscillation; in others the

acceleration of the short arcs over the long arcs varied from 3 up to as much as 9 *seconds* in 24 hours. This circumstance has led some watchmakers erroneously to conclude that isochronism is not of much importance in the timing, for, although it may not be the one and only mode of adjustment, without it a chronometer is not worthy of the name.

We shall then, until the contrary is proved to be the case, continue to regard it as necessary in certain chronometers to accelerate the short arcs for the reasons already given in articles **607**, **647** to **656**, and **674**; but, in a question that is so novel and so complicated, our hesitation will be easily understood.

On the loss in the long arcs with a theoretical balance-spring.

1446.—In the operation of springing and timing chronometers as practised at the present day, the makers *try* the terminal curves until the isochronism is such as experience has shown to be best. They take account, more or less unconsciously, of the modification caused in the progressive velocities of the vibrations by the impulse and by the centrifugal tendency, in consequence of which the bi-metallic arcs open out in the long arcs, and it generally results that the curves actually adopted in practice differ materially from the theoretical curves. This observation, which is at once deducible from a recent communication to the Academy of Sciences to be again referred to, explains many cases of want of success in applying the theoretical terminal curves. For, to realize absolute isochronism of the balance-spring, its resistance should increase in proportion to the angle of movement (a fact that can be easily verified with a detached uncut balance). But, since the compensation balance opens out in the long arcs, its moment of inertia varies, and, the elastic force of the theoretical spring not increasing in proportion to this accidental increment in the moment of inertia, it follows that the long arcs will lose on the short arcs by from 10 to 12 seconds in 24 hours.

But, as has been very properly observed, this is not, as in the preceding case (**1430**) an acceleration of the short arcs *due to the balance-spring*, but rather a *retardation in the long arcs due to the balance*, by no means the same thing. In the case we have already considered, namely that of a retarding influence that had to be avoided, it would be necessary to still further increase the accelerating effect of the balance-spring. But such

a condition, with most of the balances actually in use in marine chronometers, would render the securing of a good rate difficult; and several of our best chronometer-makers have had failures in consequence of this fact which they did not foresee.

1447.—A balance-spring of the theoretical form will not secure, in a marine chronometer, all the advantages that the profound memoirs of M. Phillips would justify us in anticipating, unless the compensating balance is more rigid than those employed at the present day.

But this is not the case as regards pocket chronometers; the balance is lighter and opens out very little or not at all, and we can thus secure isochronism in the long and short arcs without any material variations when the balance-spring is provided with the theoretical terminal curves.

It follows from the preceding considerations that the form of the compensation weights is of importance; it should be such that the air influences them in a contrary direction to the centrifugal force. Weights that cut the air more easily, for example ovals, would thus give rise to a somewhat increased retarding effect in the long arcs.

Equation to express the loss at extreme temperatures.

1448.—It has been shown in paragraph **1357** that, if a chronometer is adjusted so as to give the same daily rate at two extreme temperatures, say 0° and + 30° C, the rate will be different at any intermediate temperature, and its maximum variation will be found at the intermediate temperature + 15° C, where it will possess, relatively, a gaining rate.

The difference is not the same for all chronometers, but may be reckoned to be, on the average, equal to about *two seconds* on the daily rate. Thus, if the daily rate is + 3″ at a temperature of + 15° C, it will diminish whether the temperature rises or falls and will become + 1″ at either 0° C or + 30° C.

We are indebted to the elaborate researches of an engineer, M. Lieussou, for our knowledge of the law governing the variation in the rate, a variation which is proportional to the square of the difference of temperatures T and t.

Let m be the rate of a chronometer at any observed temperature, t; T, the temperature at which the chronometer has its maximum gaining rate; a, the daily rate at the temperature T; c, a coefficient representing the change in the rate when the temperature varies from T to T ± 1°.

Knowing the rate a, the temperature t, and the coefficient c, we obtain the rate m from the following equation:*

$$m = a - c\,(T - t)^2$$

TIMING OF ORDINARY WATCHES.

It is convenient to draw a distinction between timekeepers of novel construction, etc., which have to be adjusted for temperature and in various positions, and a watch that has merely been cleaned and requires to have its index set in the proper position. We will, then, consider these two cases in succession.

Adjusting of watches for temperature and in the horizontal and vertical positions.

1449.—The order of procedure is similar to that followed in the adjustment of marine chronometers except that less minute care is needed and we count not by fractions of a second but by two or three seconds or even more in 24 hours.

Adjustment for temperature.—A manufacturer that is desirous of producing a pattern watch with, say, a cylinder escapement, should test his new calliper at different temperatures, varying the relations between: the length of balance-spring; the weight and size of balance; the dimensions of the cylinder (on which will depend the diameter of escape-wheel) as compared with the diameter of balance, this being the one logical starting-point; the size of pivots. He will select the set of proportions that secures the least variation on changing the temperature and the position.

It will be clearly understood that by this means we only secure an approximate and spurious isochronism, as has been already observed in article **1410**, since it is the resultant of two elements of which one is irregular. That is to say, whatever inconvenience may arise from the too rapid rate of increase in the force exerted by the coiled up balance-spring in the long arcs must be practically counteracted by the increased resistance of the pivots due to the lateral pressure exerted by the balance-spring and the escape-wheel tooth. These influences are still further modified by the friction against the cylinder, a friction which is proportional to its diameter (**359**, etc.).

Lever watches are much benefitted by applying isochronal balance-springs, and, when the amplitude of oscillation of the balance does not exceed one turn or one turn and a quarter, they

* Mr. J. Hartnup, of the Bidston Observatory near Liverpool, adopts the same formula, and has for many years past employed it in tabulating the errors of marine chronometers.—Tr.

can be timed very accurately with a flat spring. If the amplitude of the oscillations is greater and we have only a limited thickness, a Breguet spring is used (**1411**).

1450.—*Ordinary adjustment for position* (pocket watches).—If the adjustment for position cannot be made sufficiently perfect by modifying the form of the ends of the pivots, the thickness or the chamfering of the jewel-holes, the inner coil of the balance-spring or its length, replacing it by another of the same strength but of greater or less length, we can have recourse to the method indicated in article **431**. (See also The Watchmaker's Handbook.)

But it must be borne in mind that the method of article **431** can only be safely employed when the extent of vibration is always either greater or less than 360°. If it were in the first instance greater and became less after a time the effects as regards loss or gain on the rate, although such as required at first, would afterwards be found to be reversed.

1451.—*Adjustment with conical pivots.*—The pivots here referred to are of the form shown at *b* (fig. 40, page 330). They should be very fine at the extremity and if the friction is excessive with the watch hanging, the jewel-hole cups being already made as deep as possible, it will suffice to slightly alter the form of the pivot with a polishing zinc and rouge without touching the fine extremity. The pivot will then become rather less conical and its surface of contact will be somewhat diminished.

If on the other hand it is advisable to increase the frictional surface when the watch is horizontal, it will only be necessary to imperceptibly shorten the pivots and make the cone rather more obtuse, diminishing it towards the point.

This form of pivot is objectionable in that: (1) it only allows of a very slight end-shake of the balance-staff, since otherwise the pivots would have too much play in the holes and one or the other pivot would shake about; (2) only a small quantity of oil can be maintained in the holes and it dries up rapidly.

1452.—*Other methods.*—Some makers have endeavoured to adjust a watch for position by more or less sloping the ends of pivots, so that, when the watch is horizontal, the resistance may be applied at the greatest possible distance from the axis.

Duchemin suggested, with a view to equalize the resistances in the two positions, to set the endstones in movable plates slightly inclined to the balance-staff, the degree of inclination being modified as required.

We shall not discuss these several methods, as their weak point, the production of a variable friction, will be evident, and they are condemned both by theory and experience.

To promptly regulate watches and clocks after repair.

1453.—This operation cannot legitimately be termed timing, since it merely consists in re-determining the length of pendulum or the position of the index for a timekeeper after it has been cleaned.

The methods indicated in articles **432—9** can be resorted to for quickly restoring the rate of a watch, and the watchmaker who adopts them from the first will find them very easy of application.

The comparison balances can be replaced by small pendulums with spring or knife-edge suspensions that have the same number of oscillations, taking care when employing them not to place them too near to the balance for fear of contact (it would be better to mount the pendulums on fine pivots, the axes carrying two opposed spiral springs); or larger sized pendulums can be used, the number of whose oscillations is an exact sub-multiple of the vibrations of the balance.

As an example, let a seconds or half-seconds pendulum be arranged so as to give a sharp distinct tick, and placed in front of a balance beating 18,000 vibrations per hour. If now the two be brought into accord, each fifth beat of the watch balance should correspond to a single vibration of the seconds pendulum or a double vibration of the half-seconds pendulum. The observer should count successively 1, 2, 3, 4, 5; 1, 2, 3, 4, 5; etc.

A clock can be promptly regulated by the same means and, since the number of oscillations is less than in a watch, the operation of counting will be proportionately easier. At the present day the escape-wheels in clocks nearly always make 120 turns in an hour; and the requisite number of oscillations is very easily calculated.

The adjustment can be effected with still greater facility by employing Guilmet's *synchrometer*, described in the *Watchmaker's Handbook*. It has the advantage of enabling us to set the balance to correspond with the period of oscillation of the pendulum, etc., before the movement is taken to pieces, without the trouble of counting the vibrations, and, after the necessary repairs have been effected, it is only needful to re-establish this agreement.

A similar instrument might be employed for adjusting watches that have to be repaired and delivered to the owners without delay.

Note on the measurement of the half-shell of a cylinder.

1454.—We have explained in article **504** the construction of a compass for measuring the half-shell of a cylinder and giving 7/12ths of the total diameter or approximately so. These data are quite sufficient for ordinary purposes, but for the benefit of those who are interested in the theoretical dimensions, and who will not be content with this degree of approximation, we would add that the proportion 21 to 36 corresponds in round numbers to 199° of the entire circumference, and the proportion 20 to 36 corresponds to about 193°. The first of these would give, after the smoothing of the surface, as nearly as possible 196°.

Those who are desirous of calculating the width of the half-shells that correspond to various angular openings and who are not accustomed to the trigonometrical methods, can easily ascertain them by the aid of a table of chords such as may be found in several works on geometry. The sine of an arc (**1291**) is equal to half the chord that subtends an arc of double the amount. (See the Watchmaker's Handbook.)

Note on the use of models for demonstration.

1454 (b).—In performing mechanical experiments the relations between the velocities should be carefully observed, but more especially is it necessary to take account of the loss of force due to impacts, etc. Thus in the experiment described in article **656**, the force exerted by the weight in virtue of gravity is 4, but its effect is only represented by 2; the difference being lost in elastic reaction, etc., and the loss is all the greater as the lever is made more elastic. Those who may be desirous of repeating our experiments or of making others similar to them should be careful not to draw absolute conclusions from them unless they are sufficiently versed in the subject to be able to take exact account of all resolution or dissipation of force, etc.

It is only by a thorough study of the laws of mechanics in its widest sense, after the manner of P. Le Roy and Graham, that the horologist of the present day can hope to make any real progress.

Concluding Remarks.

In concluding this Treatise on Modern Horology in Theory and Practice we can but express a hope that it may serve the purpose for which it is intended; namely, to place theoretical instruction together with practical results thoroughly established by experiment within the reach of manufacturers and all others interested in the subject.

But in order that it may continue to be regarded as a standard and exhaustive work of reference it must be kept abreast with the discoveries that are made in our industry as well as the changes introduced into practice.

New editions of so large a work would be objectionable in that they would diminish the value of those already in existence. The Treatise will therefore be completed by the publication from time to time of supplements, containing such additional and novel information as appears necessary to maintain the character of the work; and we would invite our readers to communicate any suggestions with reference to these supplements, which shall receive due consideration.

APPENDIX.

MISCELLANEOUS ARTICLES.

FURTHER CONSIDERATIONS ON TIMING.

To ascertain true time. Daily rate. Error. Record of rates. Mode of counting. Oven, etc.

1455.—*To ascertain true time.*—For general purposes a meridian traced out with ordinary care and tables of the equation of time will suffice; but for accurate timing the tables published in the Nautical Almanac must be consulted. If an observatory is not accessible the time of transit of the sun or a star across the meridian must be ascertained by means of a sextant or of a mural circle firmly fixed to some immovable foundation, etc.; for it is important to be able to test the invariableness of the rate of even good regulators every eight or ten days. Works on Astronomy give full explanations of the method of employing these instruments and the opticians that deal in them may be applied to.

1456.—*Limits of accuracy.*—A variation of 1 second a day in the rate of a chronometer would correspond, in 30 days, to an error of ½ a minute of longitude in time; or about 8½ miles at the equator, where the error would be of the greatest moment. Such an amount of variation is generally only the result of an accumulation of daily errors, for in astronomical regulators and good chronometers the difference between one day and another is not more than hundredths of a second. Rapid changes are very rare since such instruments are not subjected to sudden alterations of temperature or position when in use.

1457.—*Daily rate.*—The daily movement of the earth takes place in 24 hours of mean time, or 86,400 seconds. The daily rate of a chronometer would then be accurately represented by this number if it went exactly to mean time, but that could never be the case except momentarily. The timing always leaves a slight difference which may be either a gain (+) or a loss (—), and we therefore have:

$$\text{Daily rate} = 86{,}400'' \pm \text{a fraction of a second,}$$

and, if we assume there to be a gain of 0·2 second, we have,

$$\text{Daily rate} = 86{,}400 + 0{\cdot}2 \text{ seconds, or simply} = + 0{\cdot}2''$$

1458.—*Error.*—This is the difference between the time as indicated by the chronometer and that taken from the local regulator, so that the error varies with every change of longitude. Assume a chronometer to be brought to Greenwich from a place whose local time is 20 min. 3 sec. in 24 hours behind Greenwich time, we shall have the error = — 20′ 3″.

1459.—*Record of rates.*—The following represents a mode of recording the behaviour of a chronometer.

Chronometer No. 61.

Date. — 1869.	Error as compared with the regulator.	Daily Rate.	Remarks.	Thermometer.
March.				
1	+ 3′ 5″·2	. . . .	. . . .	10°·4 C
2	+ 3′ 4″·3	— 0″·9	Horizontal position.	11°·2
3	+ 3 2′·2	— 2″·1	,, in oven.	x
4	+ 3′ 3″·8	+ 1″·6	,, ,,	28°·6
5	+ 3′ 5″·0	+ 1″·2	Vertical position.	11°·8
6	+ 3′ 6″·3	+ 1″·3	,, ,,	9°·3

The mean daily rate for a number of days is obtained by adding together the several daily rates and dividing by their number, having regard to the sign (+ or —).

1460.—English chronometers that are submitted to the Greenwich trials are exposed during about six months to temperatures varying from 0° C (32° F) to 38° C (100° F), the rates in different positions as referred to the meridian being recorded in order to ensure that the instruments are not subject to magnetic influence. Their order of merit is determined by a "trial number" obtained by adding the difference between the greatest and least weekly rates to twice the greatest difference between one week and the next, double weight being given to this last source of error since it is of greater importance: it will thus be seen that the best chronometers are characterized by having the *least* trial numbers. The Admiralty then purchase such instruments as are required for the service of the Government on the results of these trials.

1461.—*Method of counting.*—Fractions of a second may be determined by counting the vibrations or the movements of the hand carried by the fourth wheel. Since the escapements of chronometers have one beat dumb, for 4, 5, and 6 vibrations a second we have 2, 2½ and 3 beats corresponding to 1/2, 4/10 and 1/3 of a second. Continue counting for example from a coincidence until a beat of the chronometer coincides a second time with one of the regulator, and this will then correspond to a difference of one second. Dividing the number so obtained by the number of seconds that have elapsed we shall determine the difference per second, etc. (See article **1453.**)

1462.—*Oven for tests at elevated temperatures.*—The watchmaker can easily arrange for himself a small oven that will suffice for nearly every case. One form is represented in fig. 9, plate XXI. (the portion from h to b' being shown in vertical section).

In a well-made wooden box $b\,b'$ about 10 or 12 inches in height and diameter, which is open at the lower end b' and closed by a lid h, is placed the box $d\,d$ fixed at its upper end in such a manner that it is surrounded by an open space $a\,a$. This inner box is closed by a lid which is fitted with a pane of glass. Within the box is placed a thermometer and a small table t, on which the watches to be tested are placed.

The outer case is closed at *z z* by a sheet of iron or zinc and below this is a coarse wooden screw passing through the cross-bar *y* and forming a support on which to place a lamp that can thus be raised or lowered at will. As we have already indicated, the lower portion of the outer case is open at the front or sides but it ought not to be exposed to air currents.

The thermometer in the interior of *d d* is so arranged that by raising the lid *h* its indications can be read through the glass in the lid *c*. Some chronometer-makers place the thermometer outside, and the tube is bent so as to bring the bulb inside the box *d d*; but the adjustment in this case requires some care and precautions which are usually unnecessary when testing watches.

It will be evident that the box *d d*, isolated in the middle of the vessel *b b'*, is completely surrounded by air which is heated by contact with the plate *z z*. If required, the interval between the two lids *c* and *h* may be filled by a cushion, and a few holes lightly closed by corks may be made in the lower portion of *a a* if any inconvenience is anticipated from dilatation, but as a rule such a precaution is not needed.

Some makers form a small oven of tin boxes inserted one in the other in a manner similar to that above described. Such an arrangement may be enough for their purposes, but it should not be forgotten that metal is a better heat-conductor than wood.

These details will enable any watchmaker to construct an oven for himself at a trifling cost; and he can modify or improve it so as to be available for the most accurate timing, since the problem to solve is merely this: to gradually produce temperatures that are more or less elevated and can be maintained constant for a sufficient length of time.

Generally an oil-lamp is employed, but for a small oven a night-light will suffice.

1463.—The *icebox for trial in the cold* is almost of the same form, but the circular space *a a* is larger so that a sufficient quantity of broken up ice can be introduced to completely surround the box *d d*, which had better be made of zinc.

REPEATER WITH FIXED STAR-PIECE.

1464.—The mechanism employed in a repeater watch is described in most works on Horology; we shall, therefore, here only draw attention to certain modifications that have been recently introduced in it. They consist in making the centre of the star stationary whereas in the ordinary repeater it is capable of a movement of recoil, and in certain changes that this modification renders necessary in the gathering rack and the "all-or-nothing" piece.

The following is a description of this new system.

The star E (fig. 1, plate XVII.), in place of being carried by the all-or-nothing piece, is mounted on a stud fixed in the plate of the watch.

The rack N, with a centre of movement at *n*, carries a finger D moving on a screw and is provided with an attached arm B which moves freely in a groove in the plate. The screw V is screwed into this arm and passes through a slightly elongated slot in the body of the rack that extends under the piece D, and a pin, fixed perpendicular to the rack, passes freely into another rectangular opening formed in the arm B ; it is indicated by faint lines at *a*. This arm can then perform a short backward and forward movement, and, when it does so, the screw V will cause the finger D to move, for the head of V rests against this finger, whereas at the opposite side (towards V) it touches the end of the slot in the body of the rack.

The piece *r* R *s* T, on which the point of the finger D rests, replaces the ordinary all-or-nothing and is termed the "spring all-or-nothing."

It is now easy to explain the mode of action of the entire mechanism.

The movement of the bolt or slide in the case will cause the rack to rotate on its centre *n* ; its teeth will wind up the mainspring of the repeater, at the same time causing the hour-ratchet C to rotate backwards until the arm B comes in contact with the snail L. The arm B will then move backwards and, since the screw V presses on D, this finger, being pivoted at its centre, will, in its turn, press against the piece *r* R *s* T, which will give way. Its extremity T in moving towards the right, will release the quarter-piece, which will fall and allow the striking to take place.

The action of the pallets *f*, *i*, *j*, of the hour-ratchet, etc., differs in no respect from that of ordinary repeaters.

We believe that the first attempts to make repeaters with fixed star-piece were those of Stagden. The form shown in plate XVII., which was kindly sent to us by M. Benoît, Director of the School of Horology at Cluses, only differs in unimportant particulars from that designed by M. C. H. Audemars.

KEYLESS WATCHES.

1465.—Ever since the invention of Beaumarchais' watch, which was wound up by rotating by the nail a ring that surrounded the dial, and that of Lepine,* where the mainspring was wound up by several times forcing the push-piece inwards after having turned it through a quarter revolution, a great number of watches have been devised in which the winding and the setting of hands is accomplished by a special mechanism adapted to the watch, situated usually in the pendant.

We shall only describe two or three of the more important or more recent

* As chief of the celebrated horological manufactory erected at Ferney by Voltaire, Lepine, who was a native of the town of Gex, very materially improved the calliper of the watches of his day. He established a house in Paris and, although he did not seek for fame and was the enemy of all charlatanism, the excellent quality of his work sufficed to secure for him two rare accompaniments of real merit ; fame and fortune.

forms, as M. A. Philippe, of the well-known Geneva firm of Patek, Philippe et Cie, has published an excellent volume on this subject, entitled *Les montres sans clefs*.

1466.—*Lecoultre's keyless mechanism.*—Figures 5 and 6 of plate XVIII. show the type of keyless mechanism in most frequent use at the present day. It is known generally as the Lecoultre mechanism, after the principal manufacturer of this form, although the design is mainly due to M. Audemars.

The winding arbor or stem is plain from a' to a' and it is shown in position fitted with the several appendages at $a\ a$ (fig. 5). A brace $i\ i$, passing into a groove that is cut round the stem, retains it in position and prevents it from falling out. On the round portion of the axis below the brace the winding pinion b rotates with easy friction and its cannon terminates in ratchet teeth so as to engage with the piece $c\ c'$ (figs. 5 and 6). This piece, which is termed the crown or set-hand wheel, can slide along the square of the arbor, and its lower end is cut with ordinary crown-wheel teeth.

The spring R, whose flexible extremity passes into a groove cut in the middle of $c\ c'$, gives certainty to the action of this wheel, while at the same time enabling it to move downwards during the backward motion of the button. We shall presently explain another function of this spring.

Winding.—When the button is caused to rotate from right to left, the piece $c\ c'$, being carried on a square, causes the pinion b to rotate through the engagement of the ratchet teeth. This pinion winds up the mainspring, transmitting its motion to the barrel arbor through a series of intermediate wheels which are partially indicated at y and x in fig. 10. On the return of the button the pinion b remains stationary, and the piece $c\ c'$ merely rotates backwards like a ratchet wheel.

Setting hands.—If a finger be pressed on the stud p, the spring R forces the crown-wheel $c\ c'$ downwards, and this, leaving the ratchet wheel b', engages with the wheel m (fig. 6). This wheel, either directly or through intermediate wheels, transmits the right or left-handed movement of the arbor $a\ a'$ to the motion work and so to the hands. The pinion b then remains stationary, for the arbor rotates within it without carrying it forward, and, when the finger releases the stud, the several pieces will return to the position shown in fig. 5.

The pivot of the winding arbor is carried in a small cock or in the thickness of the plate of the watch. At its upper end the arbor is held with slight friction in a cock or in the band of the case. The pieces $b\ b', c\ c'$ are let into a recess which is sufficiently large to ensure their proper action without contact. With such an arrangement it is possible, by merely unscrewing the brace $i\ i$, to withdraw the arbor or replace it at pleasure without the necessity of removing any other piece.

1467.—*MM. Patek and Philippe's keyless mechanism.*—At first the keyless watches manufactured by this firm were not provided with a stud for setting the hands; but subsequently they introduced the arrangement shown in fig 10, plate XVIII., which we proceed to describe.

The arbor and its accessories are shown separately in figure 11. The upper ratchet forms part of the arbor; below it is the winding pinion a, and below that the small crown-wheel f. The arbor is round, only a small portion, shown to the left of r, fig. 11, being filed flat. There is a projection in the hole passing through f that corresponds with this flat face, so that the crown-wheel is always compelled

to rotate with the arbor; the projection is formed by fixing a small pin in the body of *f*.

It will be evident that on rotating the arbor for the purpose of winding the watch, an action similar to that of a Breguet watch-key will occur as with the preceding arrangement, and, on pressing against the stud *j*, the entire system *af* will be depressed, the engagement of the crown-wheel *f* for setting hands will be effected and the hands can be rotated at will to the right or left, the pinion *a* being meanwhile out of action, since it rotates within the circle of crown teeth of the winding up wheel, some of which are seen at *x*.

1468.—*Keyless mechanism without stud of MM. Patek and Philippe.*—Fig. 12, plate XVIII., represents the several parts of this mechanism, which is of comparatively recent date, in position for effecting the winding of the mainspring. As in the ordinary system, the pinion *b* runs freely on the arbor *a* and it is maintained against the shoulder *k* of the arbor, this shoulder being formed of a separate piece of metal screwed to the arbor for a reason that will be subsequently explained. The piece *d*, which is known as the cradle, encloses within its two forked arms the shoulder *k* and the pinion *b*; the cradle rotates on the shouldered screw *d*, and is maintained stationary by the head *e* of the spring *e' e*, which presses against an inclined face of the cradle. A second spring, *f*, presses against a groove cut in the small set-hand crown-wheel, thus maintaining the engagement of the two ratchet wheels. When it is required to change the position of the hands, the arbor is drawn outwards by the button fixed at its extremity until the cradle *d* forces the spring *e* to slide along the inclined plane with which it is in contact; it reaches the edge of this plane and, on continuing the withdrawal a little farther, the spring engages with another face of the cradle in such a direction as to counteract its tendency to rotate. Figure 13 shows the several parts of the mechanism in their new positions and the set-hand wheel *s s* engaged with the motion work. The pinion *b*, notwithstanding its withdrawal, has not ceased to engage with the winding up wheel; and this fact ensures that the several parts return to their initial positions without difficulty; the least motion of the button inwards will give rise to a slight motion of the cradle and the spring will be released, bringing the two inclined faces in contact as shown in fig. 12, and, since this spring is very powerful, it suffices to bring the mechanism into position for winding up.

We mentioned that the shoulder *k* is screwed to the arbor *a*; the object of this is to facilitate the introduction and removal of the movement to or from the case. By placing a screw-driver in a notch cut across this shoulder, it can be held stationary while with the other hand the arbor is unscrewed; the arbor can then be removed and the movement is free to be taken in or out of the case.

Keyless watches that did not require the stud were prior to this produced by the firm of Breguet, and one of these is described in volume III. of the *Revue Chronométrique*.

1469.—*Observations relating to keyless mechanisms.*—The Theory of Depths affords the intelligent watchmaker ample means for securing an effective arrangement for winding up the mainspring; so that the principal difficulty consists in the mechanism for setting the hands. If the stud only projects a very short distance, the owner experiences some inconvenience in moving the hands; if, on the other hand, it is too prominent, the stud is liable to catch in the pocket and

under any accidental pressure, etc. Moreover in the arrangements in use at the present day the motion is communicated to the hands through an arbor held by friction, which will only remain effective if the adjustment be perfect, etc. The perfecting of the set-hand mechanism then should be the subject to which special attention is directed by intelligent inventors.

CHRONOSCOPES, CHRONOGRAPHS, COUNTERS AND COMPARISON INSTRUMENTS.

1470.—These several appliances are all more or less alike; they are employed for measuring small intervals of time. The name *chronoscope* is more especially applicable to those in which the movement is continuous; thus instruments governed by a conical pendulum, Wagner's differential movement, the Foucault regulator, etc., belong to this class. In several the commencement and termination of an observation are automatically recorded by electricity.

Chronograph is the name more especially applied to an instrument that measures time by a succession of intermittent movements, such as those of a horological train controlled by an escapement. Its principal characteristic is an ink-marking hand that will be presently described (**1474**). On depressing a button this hand records the several instants of the observations. The credit of this invention belongs to Rieussec, a watchmaker of Paris, who introduced it in 1822. Breguet, a few years afterwards, patented an instrument that differed from that of Rieussec only in this: the latter caused the dial to rotate while the hand remained stationary, whereas Breguet had a fixed dial and a hand that revolved. It was merely a corollary of the original design.

Lastly the term *counter* may be applied to any instrument for measuring intervals of time; thus there are simple counters, chronograph counters, etc. Their number is very great and we shall only refer to a few of the principal forms.

MM. Jacob and H. Robert's Counter.

1471.—These instruments will be described and illustrated in the Watchmaker's Handbook, to which we therefore refer the reader.

Seconds Counter of M. Winnerl.

1472.—*Simple counter.*—In this instrument the seconds hand can be stopped at will and then instantaneously brought to the position it would have occupied if its movement had not been checked.

Figure 1, plate XVIII., gives a longitudinal vertical section of the mechanism of this counter on an enlarged scale; it was invented in 1831. Figure 2 shows the seconds wheel pivot in elevation and very much magnified.

The seconds wheel A is fixed to a perforated pinion B, the smaller pivot of which is prolonged. The pivot C, that traverses the plate of the watch and usually carries the seconds hand, consists of a tube terminating in two inclined

planes *a* at the lowest points of which is cut a small notch *a'*. A rod slides in this tube and carries a small ferrule *b* so formed as to engage in the notch *a'* when the rod descends in C. A steel cock *o* is fastened to the plate and perforated with a small hole in which the pivot of the rod *d*, carrying the seconds hand *e*, can rotate freely. The rod *d* is thinned away down to the shoulder *f* in order that it may pass freely but without play through the hole in the cock *o*. It will be seen that, on raising the rod until the shoulder *f* rests against the cock, the points of the inclines *a* will pass below the lowest points of the ferrule *b* without touching it.

When it is required to arrest the seconds hand, the observer presses on a stud which, acting on the pin *g*, causes the spring *h*, fixed at one end to the plate, to suddenly rise ; the forked extremity of this spring presses the shoulder *f* against the cock ; the wheel A will then continue its movement of rotation independent of the hand, but, as soon as the pressure is removed from the stud, the spring *h* is released and presses on the ferrule *b*, the point of which will slide along the incline *a*, carrying the seconds hands with it, until held in the notch *a'*, when the two will rotate together ; the point of the spring *h* will be arrested by a tongue below the cock *o* so that t neither touches the ferrule *b* nor the shoulder *f*.

In conclusion we would only observe that the action of the piece *b* and the incline *a* is very safe and prompt, and its only objection lies in the increased height required in the case.

1473.—*Recording counter.*—M. Winnerl has suggested several arrangements which are described in vol. III. of the *Revue Chronométrique*. We quote the description of the following (fig. 4, plate XVIII.)

The pivot of the seconds wheel axis being left rather long and conical, a small piece of steel, of the form of a heart (as seen at *g*, fig. 8), is fitted to it. This piece, which is indicated at 1, fig. 4, should be placed at such a distance that the oil of the pivot cannot reach it. On the continuation of the pivot above this cam is a very light plate (2, fig. 4, and *r*, fig 3) carrying a click (3, fig. 4, and *d*, fig. 3) and its spring (*b*, fig. 3). This spring forces the click against the circumference of the heart-shaped cam.

The cannon centre of the plate (*a*, fig. 3) carries a gold seconds hand (*c*, fig. 3, and 4, fig. 4) and at the extremity of the pivot prejecting above the hand *c* (in fig. 3) is fixed a blue steel hand (5, fig. 4) with a very thin shoulder so as to admit of a slight end play of the cannon centre. The two springs 6, 6, (fig. 4) are of equal strength and they press against opposite sides of the plate so that it is maintained steady and central, thus leaving the pivot of the seconds wheel free to rotate. There is thus only the friction of the click against the rim of the cam, which is rounded and polished with care.

This being understood the action of the entire mechanism will be at once appreciated.

The click 7, which is screwed to the piece 8, terminates in two teeth at a distance apart equal to half that between two teeth of the ratchet wheel 10. By pressing the push-piece 9 the click will cause the ratchet wheel to advance by half a tooth, and the spring 11, whose extremity is also double, will steady the ratchet wheel either by one or two teeth: the teeth of this wheel thus come in succession opposite the piece 12, which, by raising the springs 6, releases the plate 2. The click 3 then slides along an incline of the cam and brings the gold hand 4 to coincide with the steel hand 5.

M. Foucher's Ink-recording Counter.

1474.—This differs from the majority of those in ordinary use mainly in the spring having a double action; it first acts edgewise, and therefore rigidly, to effect a lifting, subsequently returning to its normal position in virtue of lateral elastic force. This double action, which is both simple and certain, was, as a matter of fact, first suggested by the author.

The external appearance of this appliance is shown by figure 75.

Figure 8, plate XVIII., is a front view of the counting mechanism, and fig. 7 is a side elevation. Above the dial, whose position is indicated by *h h*, is a double hand, the first or primary hand carrying a small ink reservoir *u* at its extremity, perforated by a small opening at its centre; and the second hand, termed the *pointer*, is exceedingly light and fixed by its extremity to the main hand, while its opposite end is so bent downwards as to pass into the ink and to be exactly above the small hole in the reservoir. The least pressure on this hand *t* will cause it to pass through the ink and leave a mark on the dial.

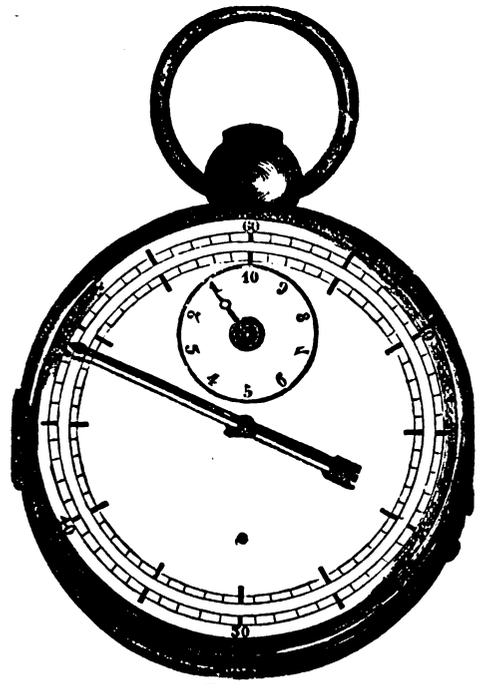

Fig. 75.

The arrangement for producing this increased pressure is as follows.

On the plate P (figs. 7 and 8) a large spring *b d*, flexible at *b* and cut away at the part that corresponds to the centre of the plate, is fixed by a screw passing through its foot. A second spring, *s n*, is fixed laterally to its free end, the elastic blade being set in a plane at right angles to the first, so that it is held edgewise to the plane of the plate. This latter spring terminates in a massive head on which are formed, at *n* and *v*, two inclined faces whose directions will be easily understood by consulting figures 7 and 8. In front of the inclined plane, *v*, is the extremity of the spring *r v*, which is flexible sideways, but rigid while retained in its own plane.

The minute hand, which is not represented, is fixed to a cannon carried on the axis *g* and to this axis is fixed a heart-shaped cam. The seconds hand is also fixed to a cannon carried on the central axis as shown by dotted lines, or on a large scale at *l* (fig. 9). To the lower end of this cannon is riveted the heart-shaped cam *c* (figs. 7 and 9). Below *c* is fixed a small spring that presses against the shoulder of the central rod, thus maintaining a friction between the cannon centre of the hand and the axis *l*, which, though slight, is sufficient.

On this cannon centre a hollow trumpet-shaped tube *a a'* rotates freely, carrying at the upper end a fork *z* (figs. 7 and 9). A fine pin is supported by this fork, passing from *z* to *z* (fig. 9), and it rests on the flexible portion of the pointing hand *t*. The tube is held up by the elastic force of this hand.

If now the push-piece be suddenly forced inwards the extremity of the spring *v* will be driven under the head *v n* of the spring *s*, and thus the piece *d s n* will be raised to the position *i i* (fig. 7). The pressure of the push-piece will bring the spring *v* to *x*, so that the head *d* of the spring *b d* will instantly fall and strike against the shoulder of the tube *a a'*. This will cause the upper hand, *t*, to be depressed and its point, passing through the ink, will make a black mark on the dial between the graduations and within a blank ring which is left for the purpose.

On removing the finger from the push-piece the spring *x r* repels the spring *n* to *f* and the two at once come to rest. The marks can be made rapidly one after another, since the hand of the operator is not so prompt in its action as the spring. If it is desired to diminish the weight of the small tube *a a'* (fig. 9), it may be made of aluminium.

The piece M, which is worked by a slide, enables us at the conclusion of the operation to bring the minute and seconds hands to zero. Its two projecting points press simultaneously on the heart-shaped cams, *c* and *g*, and cause them to rotate (since they are held only by the pressure of a spring), until these points are arrested by the points of junction of the sides of the cams.

M. Redier's Chronometrical Comparing Instrument.

1475.—The action of this appliance depends on the differential movement of the two last axes of two superposed trains of wheels.

The upper train has the escape-wheel axis at the centre D of a dial (fig. 76); this axis performs one revolution in a second and carries a small plate in which are three pins at equal intervals apart. The last axis, E, of the lower train carries a small lever F which is held stationary by one of the pins in the plate, from which it escapes three times in a second. We thus have almost precisely the same effect as in seconds watches with a double train of wheels.

The difference consists in this: The two trains are contained in a box as shown in fig. 77, so formed that its two cylindrical portions rotate one round the other, and the train of wheels in connection with D is attached to the lower portion while that connected with F is fixed to the upper cylinder. Although it has reference to another instrument, fig. 7, plate XIX., clearly indicates at F and P the manner in which the two trains are correlated.

If now such an instrument be in exact accord with a seconds regulator, it will be seen that, although no change be made in the train that drives either the disc D or the lever F, if the centre of this lever be displaced, say, from E to I (fig. 76), when the disc rotates in the direction of the arrow, the release of F will be delayed by an appreciable quantity and the beats of the instrument and the

regulator will be no longer coincident. If the centre E be moved to G, so that the displacement amounts to a third of the circumference, the coincidence will be restored, but with this remarkable difference that the instrument is a second behind the regulator. By continuing to rotate the centre E in this manner, the retardation of the instrument can be made 2, 3, 4, 5, &c., seconds.

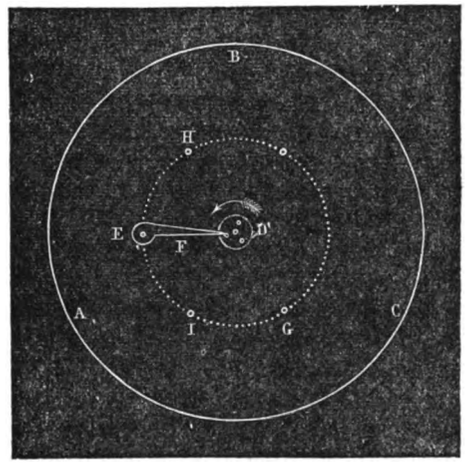

Fig. 76.

But if a displacement through one-third of the circumference occasions a loss of 1 second, a displacement of one-sixth will cause a retardation of ½ a second; in short, any given amount of displacement will occasion a proportional amount of retardation in the instrument. To make it one-tenth of a second behind the regulator, E should be moved through one-tenth of the distance from E to G; for 3/10ths it must move through 3/10ths of this distance, and so on.

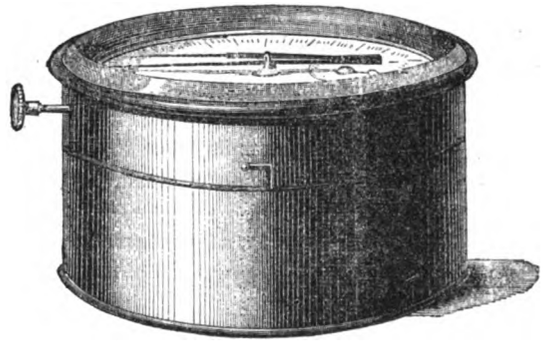

Fig. 77.

If now, instead of displacing the centre E towards G, it be moved towards H the effect will be reversed, and the instrument will gain in a manner similar to that already indicated.

Such is the principle of this appliance, which is termed by its author a chronometrical comparing instrument, because its most important use is to give an exact determination of the difference between two clocks or chronometers.

The following is a partial description of its details of construction:

Fig. 77 is an external view of the instrument.

The dial has three series of divisions. The one concentric with the case serves to measure the movement of the seconds hand (x, fig. 4, plate XIX.). An excentric graduation indicates the hours and minutes as in an ordinary watch (M, fig. 4); and the movement of the hand (I, fig. 4) over the third series of subdivisions gives the angle traversed by the centre E of the lever in fig. 76.

We have observed that the case is divided into two portions one above the other. The lower one contains the train by which motion is communicated to the disc D, and in the upper portion is the seconds train connected with the lever F. When it is required to set the instrument so that it coincides with a chronometer, the upper portion is rotated to the right or left until the exact accord is established. If the instrument differs from the clock by 25·7 seconds, it will be necessary to make eight complete revolutions of the upper case to give 24 seconds, one-third of a revolution for the 25th second and 7/30th for the 7/10ths second.

The following arrangement is adopted by the author as a simple means of measuring the amount of this displacement:

A wheel is fixed to the plate of the lower movement and concentric with it. Another wheel of the same number of teeth engages with the first, being pivoted in the frame of the upper movement. The hand I (fig. 4) is carried on the axis of this second wheel, so that with each complete rotation of the upper case the hand I will perform an entire turn.

The hand thus records all the relative movements of the two trains and if its scale is divided into 30, each subdivision will correspond to a tenth of a second. This completes the description of the instrument in its simpler form. But to facilitate the reading off of the amounts of displacement it is well to bring the hand I to zero before each new observation; and for this purpose a cannon having a heart-shaped cam is held by friction on its axis. A push-piece P on being drawn out moves the large piece of metal M E, and the projection E, pressing against the cam, brings the hand to zero.

Two objections have been urged against the use of this instrument: the difficulty of timing with a 36,000 train, and the insufficiency of even this number of vibrations for securing a practically continuous movement of the disc.

To these criticisms it may be replied: Moinet in his thirds counter had not only ten vibrations per second but sixty, and he asserts that a good rate was secured; more recently MM. Rozé have employed a Redier Comparison Instrument with a 72,000 train which they state gave satisfactory results.

Brocot's Perpetual Calendar.

1476.—Of the many instruments of this character we will select for description the most recent arrangement adopted by M. A. Brocot who has acquired considerable reputation in this interesting speciality, a reputation which he has increased by the publication of an important volume entitled: *Calcul des rouages par approximation.*

The several parts of the mechanism are divided between the two sides of a single plate, whose front face, towards the dial, is shown in fig. 2 (plate XX.), while the reverse side is represented in fig. 1; this latter figure gives the principal parts of the mechanism for indicating the day of the week and month.

The four dotted lines in two parallel pairs join the extremities of the same axis. It should be observed that the dimensions of the plate rendered it necessary to draw the dial, shown in figure 4, smaller than it should be.

The axis A′ (fig. 2), which is identical with A in fig. 1, carries the hand for indicating the day of month (fig. 4); the axis B′ (same as B) that for the day of week, and, lastly, the axis C′ (same as C) carries the month hand. The disc Z (fig. 4) is centred at L, and the central axis carries the long or mean time hand.

The week wheel of 7 teeth and the month wheel of 31 teeth are maintained in their position by two rollers free to move on shouldered screws and resting in the interval between two teeth. They are carried on arms held up by springs.

A change in the month date is effected by the motion of the long detent M *m*, which is acted on by a pin *c* set in a wheel of the going or striking train of the clock; a wheel which of course must only make one revolution in 24 hours. The detent M carries two long clicks H and G, fig. 3, which are maintained in contact with their respective wheels, the first by a spring and the second by a counterpoise weight at its opposite extremity. The whole of this arrangement is shown separately in fig. 3.

When the detent M *m* is deflected in the direction of the arrow, by the pressure of the pin *c*, the clicks H and G will each slide over a tooth and their heads will fall into the next succeeding spaces. So that, immediately on the detent being released from the pin *c* and falling by its own weight aided by the pressure of a light spring, the clicks will cause the two wheels of 31 and 7 teeth each to advance one tooth.

Since all the months have not 31 days it is necessary that the month-day click should advance by 1, 2 or 3 additional teeth in the months with 30, 29 or 28 days respectively.

The axis of the month-day wheel, which makes one revolution in a month, carries a pinion of 10 leaves (fig. 2) which engages through an intermediate wheel D with a large wheel of 120 teeth termed the year wheel from the fact that it only performs one revolution in a year. Its axis C′ (fig. 2) carries the hand that points out the month, the *equation cam* which we shall not consider at present, and at the opposite extremity, C (fig. 1), a small wheel that drives another much larger wheel with four times the number of teeth; so that this latter wheel performs a complete revolution in four years. The disc V F is fixed to its surface so that the wheel itself is not visible except through the spaces to the number of 20 that are cut in the circumference of V F, this being the number of months in four years that have not the full number of thirty-one days.

The uncut portions of the circumference correspond to the 31-day months; the 16 shallow spaces to 30-day months; the deeper space *b* to the month of February in leap-year; and the three others numbered 1, 2, 3, which are still deeper than *b* to the ordinary month of February.

In the surface of the month-day wheel is a pin *i* which should be set opposite to the tooth that corresponds to the 28th day of the month. On the detent M *m* is pivoted an angular piece N *n*, with a pin *o* projecting from the extremity of its vertical arm *n*. Owing to the weight of N this pin rests against the circumference of the disc V so that it will drop into the spaces whenever they are brought opposite to it. It follows that the degree of elevation of the arm N corresponds with the position of the pin *o*, and it will descend during the motion of the detent M *m* from right to left since the pin *o* always remains in contact with the circumference of the disc or with the bottom of a space.

If the pin *o* is resting against the circumference of V, the arm N, which is then at its greatest elevation, will have no effect; its position in this case is indicated by dotted lines. But if the pin is at the bottom of a 30-day space, the

horizontal arm N will descend so that, during two-thirds of the advance of the detent, it engages with the pin *i* in the 31-toothed wheel, and will advance it one tooth during the last third of the course of M *m*; in other words it will cause the hand to advance from 30 to 31. The sudden release of the detent by the pin *c* will again advance the hand to 1.

At the end of February in leap-year the arm N engages at the first third of the path of the detent and causes two teeth to pass. The return of the detent will advance to the 1st day of the month.

For the ordinary month of February the arm N engages immediately on the motion of the detent commencing and advances the wheel by three teeth, etc.

On examining the figure it will be seen that one side of each space is inclined in order to facilitate the exit of the pin *o*. It is released from the deep notches by the finger J, fixed to the axis of the year wheel. Since the angular movement of this finger is much more rapid than that of the disc V, it will be brought opposite each deep notch and raise the pin from it before the position of the disc has materially changed; it will moreover hold the pin up long enough to ensure its falling into a suitable position.

The *phases of the moon* are shown by the wheel L (fig. 2) which is driven as follows. At the extremity B′ of the week-day wheel axis is fixed a 10-leaved pinion engaging with a wheel of 84 teeth. This wheel is rigidly connected with another of 75 teeth which engages with the large wheel L of 113 teeth, and on this are painted three discs (Z, fig. 4) of the same colour as the sky is represented on the dial. (The rack R does not engage with the wheel T or with the larger one below it; it merely passes between the two without contact.) With the above numbers a revolution of the wheel L will be completed in very nearly three lunar months, the error being only 0·00008 of a day per month.

Equation of Time.—On the axis of the year wheel is fixed a plate or *equation cam*, improperly term the *ellipse*, of the form indicated by the line *y s*. A pin in the rack R (which may carry a small roller) rests on the circumference of this disc, and the rack engages with the wheel K whose axis projects through the dial and carries the long central or mean time hand.

This hand should be directed towards the XII on the following days, when the mean and apparent times coincide: 15th April, 13th June, 31st August, and 24th December. On these days the pin *s* rests opposite one of the four black dots marked in the drawing (fig. 2). The graduations above the title *Temps moyen à midi vrai* (mean time at apparent noon) indicate the number of minutes that a well regulated clock should be before (+) or behind (−) solar time.

1477.—*To draw this equation cam.*—The reader will doubtless look for some general details as to the mode of drawing this approximately elliptical figure; the following plan may be adopted. Remove the pin *s* from the rack and replace it temporarily by a small pointer held by a spring above the plate from which the cam is to be formed. This plate should be of the same size as the uncut portion of the year wheel. A temporary click-spring maintains this wheel stationary and if a spring be made to press against the wheel K it will rotate with friction. The subdivision of the scale into 15 degrees on either side of the XII is determined by the amount of displacement of the hand when the greatest radius of the cam is in action. The mean time hand is now brought over the XII and a mark made on the disc opposite one of the dates indicated above at which mean and apparent time coincide. Commencing from this date cause the year wheel to advance by one tooth at a time (in the present case each tooth

corresponds to three days); by the aid of an equation of time table set the hand opposite the number of divisions corresponding to the date indicated; mark another point, and so on round the entire circumference of the disc.

Draw a curve through the series of points thus obtained and cut the disc to it, leaving a slight excess of metal. Set the pin *s* in position and it will then be easy to observe the indications, modifying the form of the cam if necessary.

CONICAL PENDULUM.

Various Applications and Experiments.

1478.—The conical pendulum, while it has not given entire satisfaction to those that required a perfectly uniform rate, has nevertheless many interesting applications; we shall, however, only make a few observations on them.

1479.—The question as to the best mode of securing a uniformly continuous movement has been much studied but it does not yet seem to be near a complete solution. The conical pendulum affords a very approximate solution. The ingenious instrument of which the first specimen was exhibited by M. Wagner at Paris gives good results, but it is influenced by one source of error which we proceed to indicate.

A (fig. 1, plate XIX.) is the seconds wheel of an ordinary clock; B, the seconds wheel of a train controlled by a fly; *c*, a roller intermediate between the wheels A and B, that is mounted in a specially formed frame capable of oscillating from right to left. If A moves slower than B the roller *c* will move towards *c'* and conversely; it is this displacement of *c* towards *c'* that is employed as a means of modifying the speed of the governor of B to correspond with the mean velocity of A. If this displacement is maintained it indicates precisely the amount by which the continuous and step-by-step hands disagree; an error that may amount to some seconds.

When a conical pendulum is employed the differences are not more than fractions of seconds. These differences are due to the fact that it is almost impossible to ensure that the bob shall describe perfect circles.

It can only describe ellipses that approximate very closely to circles.

The length is measured in the same manner as that already explained (**1282**), and the rate maintained by it, while inferior to that with an ordinary pendulum, is very satisfactory.

Between the two periods of a pendulum, whether ordinary or conical, that measures 994 mm. (39·13 ins.), coming to its starting point, an interval of 2 seconds occurs, and, if of any other length, the two kinds will give equal velocities.

M. Balliman adopts a very simple method of driving the instrument and suspending the pendulum. A metallic wire supports the rod of the latter, or it may reach the entire length of the pendulum, and this is all that is needed; it is sufficiently fine to bend at the point of suspension and there is no occasion to fear the influence of torsion, since the pendulum remains always in the same plane.

MM. Cuel and Rozé have proposed a form of suspension with 4 laminæ, which is shown in fig. 2, plate XIX., in front and side elevation. It is merely one suspension supported by another at right angles to it.

Figure 5 represents a clock with this suspension and the pendulum is driven in the manner suggested by M. Balliman.

M. Redier has taken advantage of this form of pendulum to arrange for advancing the hands or setting them back by a given quantity without stopping the clock. L is the arm that drives the pendulum P and it makes a revolution in every two seconds.

Assume the clock movement to be horizontal. It is evident that if this movement be caused to rotate by means of the handle in the same direction as the pendulum and with the same velocity, the hands will remain stationary.

But if such a movement be only continued for a short period, the amount will be subtracted from the motion of the hands or added to it according as the motion is in the direction of the pendulum or the reverse.

This is the principle of the comparing instrument already described (**1475**), and it has been taken advantage of by M. Redier as affording a means of electrically transmitting the time to any required degree of accuracy, by adding or subtracting certain definite periods.

The clock represented in fig. 6 produces the same effect by the mere displacement of certain wheels of the motion-work.

The right-hand dial records the continuous seconds and that to the left is similar to an independent seconds watch.

Figure 7 indicates the manner in which these several functions are performed; *b* and *c*, engaging with *a* and *d*, are carried in a frame Y Y U U to which a circular motion can be given.

By displacing *b* and *c* by means of the plate wheel Y Y the instant at which the arm F engages with the pins P is varied and, as in the comparing instrument, M. Redier succeeds by this means in securing the required coincidences.

1480.—Only by employing a conical pendulum is it possible to obtain a clock that indicates at the same time mean and sidereal time, the differences in fractions of a second being legible on the two dials.

Figure 3 represents a portion of the train of such a clock in outline. If M is the mean time seconds wheel, and S the sidereal time seconds wheel, by adopting the train $\frac{23 \cdot 58 \cdot 79}{38 \cdot 57 \cdot 70}$ from M to S, we obtain the proportion $\frac{24201}{23270}$ and this is correct within a yearly error of 2·43 seconds.

1481.—We would refer to an exceedingly curious experiment with the conical pendulum.

It is known from the celebrated experiment of Foucault that the ordinary pendulum will remain in its plane of oscillation, however the point of support is moved. This physicist has by this means demonstrated the rotation of the earth on its axis.

The conical pendulum is characterized by the same property, and to prove that this is the case M. Redier has arranged a small clock with conical pendulum on a rotating plate concentric with the pendulum. When the entire clock is rotated by means of this plate, the motion of the pendulum not being modified by this displacement, the seconds hand will be observed to lose by an amount that is proportional to the movement of the plate when this takes place in the same direction as that of the pendulum. The hand will, on the contrary, gain in a similar proportion if the plate and pendulum rotate in opposite directions.

Thus the clock can be caused to lose or gain without touching any portion of it, but merely by rotating the entire mechanism round its axis.

VISSIÈRE'S MERCURIAL COMPENSATION PENDULUM.

Supplementary notes.

1482.—Fig. 1, plate XXI., represents the form of mercurial pendulum adopted by this well-known maker. It is half the actual size.

Figs. 2, 3, 4 and 5 show details of construction.

Fig. 2 is the steel pendulum rod detached.

Fig. 3 is the stirrup *b a a* seen from above.

Fig. 4 is a double collar for steadying the upper ends of the glass tubes.

Fig. 5 is a section of the timing screw showing details of its construction.

These several figures hardly require explanation.

1483.—*Notes.*—The mercury tubes can be made of cast iron but it is important to remember that the expansion will then be greater in all directions.

The dilatation of glass is generally irregular (**1265**), but this circumstance has very slight effect, and it is usual to prefer it to iron. For glass permits of a slight reduction in the height of the mercury column and it is possible to examine the column and to observe any air-bubbles or impurities.

The mercury should be very pure and free from air. Makers who undertake the construction of this class of pendulum should make themselves acquainted with the best methods of purifying the metal.

A first approximation to the required dimensions may be obtained by calculation, taking the coefficients of dilatation of the several parts as a basis; but in order to ensure absolute accuracy it is the safest plan to employ an oven arranged for testing purposes.

INSTRUMENTS FOR MEASURING THICKNESS, &c.—MICROMETERS.

Ordinary gauges.

1484.—The instruments for gauging in which an index is moved by the agency of a train of wheels, those in the form of a proportional compass that are known as douzième (or twelfth of a line) gauges, and compasses with index similar to that shown in fig. 4, plate XVI., are objectionable in that the opening of the jaws gives a measure of a *chord* whereas the displacement of the index measures an *arc* of a circle; and these two quantities do not progress in the same proportion. It follows from this that if the index is first arrested when pointing to 15, for example, and again when at 30, the interval between the jaws in the second case will not be exactly double the first.

The watchmaker can, however, render the numerical proportion between these two numbers absolutely correct.

We will first assume that the two jaws (*n*, fig. 4) are accurately joined throughout their entire width when the point of the index is superposed upon the line *i c*; insert between them a small perfectly round cylinder, say, two millimetres in diameter, so that the point of the index is brought to *a* on the line *i a*; we shall then have the angle *a i c* equal to *a* D *c*, fig. 9. Place a carefully graduated scale on the metallic plate in the manner indicated in fig. 9, set one point of a compass in D (the axis round which the finger *i c* rotates), and by trial ascertain the radius for describing the arc *a b c*, such that its chord (or the straight line *a c* joining the two points of intersection of the arc with the lines D *a*, D *c*) includes exactly 40 graduations, say millimetres, on the scale. Then fix the scale in this position by screws or sealing-wax, etc.

From the centre *c* and with a solid pair of compasses with fine hardened points, draw from each graduation of the rule a small arc to meet the arc *a b c*, the eye-glass being maintained at the eye during the process. Make a mark at each point of intersection.

When the circular arc is thus graduated spaces traversed along it by the extremity of the index (whose length must be equal to *i a*) will always be proportional to the successive distances between the jaws.

The gauge represented in fig. 4 will measure to the twentieth of a millimetre. If the lines *i a*, *i c* had been of double the length, 80 millimetres would have been included between the points *c* and *a*, so that the jaws *n* would then measure to the fortieth of a millimetre.

Various kinds of graduation can be obtained with the same scale by taking 1, 2 or 3, &c., of its divisions as unity.

The watchmaker will find an advantage in possessing another gauge terminating with points (K, fig. 4) instead of jaws, but when using it he must not forget that, if placed in a hole, the point will not correspond with its centre, in consequence of the fact that the axis is inclined.

Sliding vernier compass.

1485.—The principle of the *vernier* is explained in the "Watchmaker's Handbook."

Fig. 8 (plate XVI.) represents this compass, and it consists of a rectangular steel rule *b a*, having a projection at right angles at *a* which terminates in a point. A slide *f d* can be moved along this rule, having a corresponding projection, *c*, at right angles to itself. The two opposed faces of these projections accurately coincide throughout their entire length, and the extremity of each is formed into a hemi-cylinder. When the two are in contact, as shown in the figure, they may form a pointed cone when points are needed, or they may be sharp-edged jaws, like those of pincers. As has been already mentioned, the slide *f d c* moves along the rule *b*; the block *s*, which is connected with the slide by an adjusting screw, will also travel along the rule with it, the clamping screw being of course released.

The rule is usually graduated in millimetres. The vernier, measuring 19 millimetres, is divided into 20 equal parts. It follows therefore that if the slide is displaced from the zero position to the point at which the first division of the vernier corresponds with the first division of the rule, the jaws will be separated by an interval of *one twentieth* of a millimetre; with the second division of the vernier and second of the rule the interval is *two twentieths* and so on up to the point at which the two twenties coincide. When such is the case the interval is an exact millimetre.

It will be at once evident that the instrument only requires to be opened to the approximate diameter of the object that has to be measured, the block *s* fixed by the clamping screw, and the jaws set to the exact distance apart by the adjusting screw. The rule itself will give the number of millimetres and the vernier the additional fraction in twentieths of a millimetre.

If well made this gauge is more exact than a gauge provided with an index finger, since there is of necessity always a slight play at the pivots in this latter; but unfortunately the great majority of the sliding compasses to be met with in

commerce are badly constructed and inaccurate. The one here described is entirely of steel with the exception of the sliding block *s*; the adjustments are very easily effected so that the instrument may be made of great service.

Micrometers.

1486.—*Duchemin's Micrometer.*—The two poppets A and B (fig. 78) support two semi-cylindrical centres. The inner extremities of these centres are sloped off to an edge and come together like the jaws of a parallel vice.

The centre C is fixed by a plate and screws. The centre D can be moved lengthwise with easy friction, being held in position also by a plate and screws.

The screw H which is tapped into the body of the tool at N causes the centre D to advance. The thread of H is fine and the motion is brought about by rotating the disc P, round the circumference of which are 50 notches. On rotating the disc the head of the spring R is raised from the notches, but it drops successively into these notches, thus steadying the disc when the hand is removed.

A complete rotation of P will cause the centre D to advance by one thread of the screw H, or one-fiftieth of this amount for each subdivision of the disc. A pin set near the circumference of the disc serves to indicate the point marked 50 or 0 and strikes against a projection carried by the spring R.

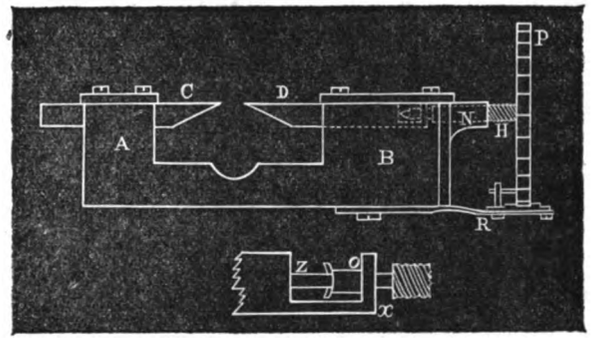

Fig. 78.

In using the instrument the centre D is advanced until the pin in the divided disc strikes against the projection of R, when the head of the spring should rest firmly in one of the notches. Advance the centre C forward until it is in contact with D; and, when the two exactly coincide, fix C by its screws. Separate D from C and then screw it back again until the object to be measured passes without play between the two jaws. Then again advance the movable centre until the two are in contact, counting the turns and fraction of a turn of P; this will give the required dimension.

The concealed extremity of D is cut away as indicated at Z *o*, and the end of the adjusting screw is diminished to a kind of pivot. This pivot passes through the shoulder *o x* and rests against Z, so as to push the centre forward. To prevent there being a period of rest a small collet is set on the pivot and held in position by a pin; it presses against *o x* whereas the end Z of the pivot is in contact with the opposite face.

1487.—*Saunier's Micrometer.*—Screw micrometers have two faults: (1) There is always a *backlash* when a forward is changed for a backward movement or *vice versâ*; (2) When an object is held between the jaws there is danger of an error of several divisions of the divided head, due to the elasticity of the object measured or of the instrument itself. The following is a description of a micrometer which we have designed with a view to avoid these two sources of error.

The centre *c* (fig. 6, plate XVI.) is accurately fitted in its poppet but with easy friction. It is traversed by a pin which permits of a longitudinal movement of the centre to the extent of the slot in the poppet that receives the pin. The centre C is forced inwards by the spring *r*, and the index *f*, resting with its short arm against the pin, is directed to the 0 of its scale when the pin is against the extremity of the slot.

The other centre *d d'* slides with friction, being moved by a micrometer screw that traverses an enlargement formed in *d d'*. The small shaded index corresponds to 0 on the millimetre scale engraved on the poppet head when the two jaws are exactly in contact. The pitch of the screw is 1 millimetre and, since the divided plate T has 50 notches, fiftieths of a millimetre can be measured; in other words, whatever the pitch of the screw, it can be subdivided into fifty parts. The small rod *i* serves as a guide to guarantee the free movement of the centre, and the action of the spring *n* has already been explained.

The strength of the spring *r* should be no more than is necessary to ensure that, for example, a staff will be held up when its pivot is placed between the two jaws; with a very light spring the slightest pressures will be detected.

When the object to be measured is placed against the jaw *c* and touched by the jaw *d*, the screw is advanced a little further in order to allow for the backlash in the return movement; the index *f* will be displaced and will advance through a certain number of degrees. The screw is then turned back until the index *f* returns to its zero. The small shaded index will then indicate the number of entire millimetres while the divided head T gives the additional fraction in fiftieths. It may be found more convenient in practice to let the centre *c* rest free against the spring *r*, so that, when the excess movement is given to *d* and the object has been removed from between the centres, the centre *c* can be pressed backwards with the nail, the object to be measured placed between *c* and *d*, and only the return movement of *d* utilized.

A micrometer similar to that shown in fig. 6 but with the two jaws bent at right angles, would be found very convenient for measuring the width of the leaves of pinions, the teeth of wheels, &c.

1488.—*Observations and details of construction.*—A very accurate micrometer screw is exceedingly difficult of construction. The watchmaker who is desirous of possessing an instrument of the highest excellence should apply to the makers of scientific instruments for such screws; but with care he can himself make one that is sufficiently accurate for ordinary purposes.

The very best cast steel should be used, as it is the most homogeneous attainable. Movable dies should be employed for cutting the thread in preference to the ordinary screwplate. Very little metal must be removed at a time and it is a good plan to employ two sets of dies, the first of which leaves the screw a trifle thick, while the second reduces it to the exact dimensions. The dies should be thick; in other words they should include a fair number of threads that are carefully finished with good cutting edges, etc.

To ensure that the screw is a good fit proceed as follows:

Having cut a satisfactory thread on the screw prepare a second somewhat thinner and employ this latter to cut the internal thread in the projection of the centre *d d'*; then make a longitudinal cut with a file half way across the hole and mount the one half in the tool shown at A, fig. 6, plate XVI.; that is to say the piece *z z'* rests on two rollers *t t* that can be elevated as required. The screw *v* is provided with a small handle and on rotating it the two screws are worked into each other, gradually bringing them nearer together, until the movement is smooth and uniform; the rollers *t t* are then permanently clamped. The thickness of the bar should be such as to ensure there being no bending.

The arrangement shown at B, fig. 6, plate XVI., seems preferable to that adopted by Duchemin (*z o x*, fig. 78, page 811); the drawing does not seem to require any description.

Note on Constant Force or Remontoire Escapements.

1489.—We have not space to discuss the question of constant force but propose shortly to publish an exhaustive work on it; we would, however, endeavour to convince those seekers after that attractive ideal, a constant force escapement suitable for high-class horological mechanism, that their researches are utterly futile.

Proposition.—Constant force, as it is understood by the majority of watchmakers, *does not and cannot* exist.

The several parts of the escapements of this class are very nearly always being displaced, rapidly and while subjected to slight forces; and the inventors have not studied their effects, which of necessity are very delicate, except with the aid of a merely elementary knowledge of mechanics, in fact no more than the simple theory of the lever. They have either been unacquainted with or failed to take account of the influence of certain physical phenomena, which as a matter of fact are of such importance that the mechanical effects cannot but be unstable.

For the elastic force of a spring varies with every change of temperature.

All bodies are in a continuous movement owing to expansion and contraction, and this motion occurs in starts *which do not all occupy the same period of time*. It has been shown that *affinity acts non-continuously*.

Knowing these facts to have been discovered and demonstrated by science, it will be understood that the molecules perform their movements or orbits while subjected to greater or less retardation according to the pressures and the nature of the surfaces; and, further, if a certain effect is delayed it will nevertheless ultimately be produced but with increased violence. We have evidence of this in the fact that a slight disturbance caused by gently striking a compensation pendulum will cause expansion to take place suddenly.

Moreover the adhesion between clean dry surfaces, which seems to be a form of chemical affinity, becomes greater as the surfaces are in more intimate contact, so to speak molecule to molecule. Adhesion, which is perceptible and variable, has been proved to occur at a point where impacts are frequently repeated. When subjected to very great pressure matter becomes changed and as it were passive. The condition of oil varies with temperature, time, etc.

Even ignoring the effects of capillarity and electric action, of which very little is yet known, it is evident that the separation of acting parts, the movements of surfaces in contact, etc., are always accompanied by resistances that are variable and relatively intense, as compared with the very slight forces that impel the several organs. Hence it follows that a resistance that would be as it were drowned in the greater power of a Graham dead-beat escapement, will have its influence on the going of a constant force escapement; the latter is in fact much more sensitive than the former.

This is why no remontoire escapement has given results that are better than those attainable with a chronometer or Graham escapement. Whenever the constant force has given equally satisfactory results it has only been temporary, and could not be permanently maintained.

We would add, what all watchmakers know, that the application of most systems of remontoire to a horological train renders it necessary that the wheels revolve with greater rapidity and there is thus a reduction in the period that they will continue to go; and we would conclude by reminding the reader that a distinction must be drawn between the chimerical search after forces that are absolutely constant and the invention of escapements that are intended to counteract more or less perfectly the inconveniences that arise from inequalities in the motive power of clocks or of the pull of the mainspring in ordinary timepieces. In this latter there is a wide field of research, providing always that we can prescribe logical limits, which it appears useless to precisely define.

TABLE OF LENGTHS

OF

THE SIMPLE PENDULUM.

TABLE SHOWING THE LENGTH OF A SIMPLE PENDULUM

That performs in one hour any given number of oscillations, from 1 to 20,000, and the variation in this length that will occasion a difference of 1 minute in 24 hours.

CALCULATED BY E. GOURDIN.

Number of oscillations per hour.	Length in millimetres.	Variation in length for 1 minute in 24 hours in millimetres.	Number of oscillations per hour.	Length in millimetres.	Variation in length for 1 minute in 24 hours in millimetres.	Number of oscillations per hour.	Length in millimetres.	Variation in length for 1 minute in 24 hours in millimetres.
20,000	32.2	0.04	13,200	73.9	0.10	8,200	191.5	0.26
19,000	35.7	0.05	13,100	75.1	0.10	8,100	195.3	0.27
18,000	39.8	0.05	13,000	76.2	0.10	8,000	201 3	0.27
17,900	40.2	0.06	12,900	77.4	0.11	7,900	206.4	0.28
17,800	40.7	0.06	12,800	78.6	0.11	7,800	211.7	0.29
17,700	41.1	0.06	12,700	79.9	0.11	7,700	217.2	0.30
17,600	41.6	0.06	12,600	81.1	0.11	7,600	223.0	0.30
17,500	42.1	0.06	12,500	82.4	0.11	7,500	229.0	0.31
17,400	42.4	0.06	12,400	83.8	0.11	7,400	235.2	0.32
17,300	43.0	0.06	12,300	85.1	0.12	7,300	241 7	0.33
17,200	43.5	0.06	12,200	86.5	0.12	7,200	248.5	0.34
17,100	44.0	0.06	12,100	88.0	0.12	7,100	255.5	0.35
17,000	44.6	0.06	12,000	89.5	0.12	7,000	262.9	0.36
16,900	45.1	0.06	11,900	91.0	0.12	6,900	270.5	0.37
16,800	45.7	0.06	11,800	92.5	0.13	6,800	278.6	0.38
16,700	46.2	0.06	11,700	94.1	0.13	6,700	286.9	0.39
16,600	46.7	0.07	11,600	95.7	0.13	6,600	295.7	0.40
16,500	47.3	0.07	11,500	97 4	0.13	6,500	304.9	0.41
16,400	47.9	0.07	11.400	99.1	0.13	6,400	314.5	0.43
16.300	48.5	0.07	11,300	100.9	0.14	6,300	324 5	0.44
16,200	49.1	0.07	11,200	102.7	0.14	6,200	335.1	0.46
16,100	49.7	0.07	11,100	104.5	0.14	6,100	346.2	0.47
16,000	50.0	0.07	11,000	106.5	0.14	6,000	357.8	0.48
15,900	51.0	0.07	10,900	108.4	0.15	5,900	370.0	0.50
15,800	51.6	0.07	10,800	110.5	0.15	5,800	382.9	0.52
15,700	52.3	0.07	10,700	112.5	0.15	5,700	396.4	0.54
15,600	52.9	0.07	10,600	114.6	0.16	5,600	410.7	0.56
15,500	53.6	0.07	10,500	116.8	0.16	5,500	425.8	0.58
15,400	54.3	0.08	10,400	119.1	0.16	5,400	440.1	0.60
15,300	55.0	0.08	10,300	121.4	0.17	5,300	458.5	0.62
15,200	55.7	0.08	10,200	123.8	0.17	5,200	476.3	0.65
15,100	56.5	0.08	10,100	126.3	0 17	5,100	495.2	0.67
15,000	57.3	0.08	10,000	128.8	0.18	5,000	515.2	0.70
14,900	58.0	0.08	9,900	131.4	0.18	4,900	536.5	0.73
14,800	58.8	0.08	9,800	134.1	0.18	4,800	559.1	0.76
14,700	59.6	0.08	9,700	136.9	0.19	4,700	583.1	0.79
14,600	60.4	0.08	9,600	139.8	0 19	4,600	608.7	0.83
14,500	61.3	0.08	9,500	142.7	0.19	4,500	636.1	0.86
14,400	62.1	0.09	9,400	145.8	0.20	4,400	665.3	0.90
14,300	63.0	0.09	9,300	148.9	0.20	4,300	696.7	0.95
14,200	63.9	0.09	9,200	152.2	0.21	4,200	730.2	0.99
14,100	64.8	0.09	9,100	155.5	0.21	4,100	766.2	1.04
14,000	65.7	0.09	9,000	159.0	0.22	4,000	805.0	1.09
13,900	66.7	0.09	8,900	162 6	0.22	3,950	825.5	1.12
13,800	67.6	0.09	8,800	166.3	0.23	3,900	846.8	1.15
13,700	68.6	0.09	8,700	170.2	0.23	3,850	869.0	1.18
13,600	69.6	0.09	8,600	173.7	0.24	3,800	892.0	1.21
13,500	70.7	0.09	8,500	178.3	0.24	3,750	915.9	1.25
13,400	71.7	0.10	8,400	182.5	0.25	3,700	940.1	1.28
13,300	72.8	0.10	8,300	187.0	0.25	3,650	966.8	1.31

TABLE OF THE LENGTH OF A SIMPLE PENDULUM—(*continued*).

Number of oscillations per hour.	Length in metres.	To produce in 24 hours 1 minute		Number of oscillations per hour.	Length in metres.	To produce in 24 hours 1 minute	
		loss, lengthen by	gain, shorten by			loss, lengthen by	gain, shorten by
		Millim.	Millim.			Metres.	Metres.
3,600	0.9939	1.38	1.32	1,900	3.568	0.0050	0.0048
3,550	1.0221	1.42	1.36	1,800	3.975	0.0055	0.0053
3,500	1.0515	1.46	1.40	1,700	4.457	0.0062	0.0059
3,450	1.0822	1.50	1.44	1,600	5.031	0.0070	0.0067
3,400	1.1143	1.55	1.48	1,500	5.725	0.0080	0.0076
3,350	1.1477	1.60	1.53	1,400	6.572	0.0091	0.0087
3,300	1.1828	1.64	1.57	1,300	7.622	0.0106	0 0101
3,250	1.2194	1.69	1.62	1,200	8.945	0.0124	0.0119
3,200	1.2578	1.75	1.67	1,100	10.645	0.0148	0.0142
3,150	1.2981	1.80	1.73	1,000	12.880	0.0179	0.0171
3,100	1.3403	1.86	1.78	900	15.902	0.0221	0.0211
3,050	1.3846	1.93	1.84	800	20.126	0.0280	0.0268
3,000	1.4312	1.99	1.90	700	26.287	0.0365	0.0350
2,900	1.5316	2.13	2.04	600	35.779	0.0497	0.0476
2,800	1.6429	2.28	2.18	500	51.521	0.0716	0.0685
2,700	1.7669	2.46	2.35	400	80.502	0.1119	0.1071
2,600	1.9054	2.65	2.53	300	143.115	0.1989	0.1903
2,500	2.0609	2.87	2.74	200	322.008	0.4476	0.4282
2,400	2.2362	3.11	2.97	100	1,288.034	1.7904	1.7131
2,300	2.4349	3.38	3.24	60	3,577.871	4.9732	4.7586
2,200	2.6612	3.70	3.54	50	5,152.135	7.1613	6.8521
2,100	2.9207	4.06	3.88	1	12,880,337m.930	17,903m.6700	17,130m.8500
2,000	3.2201	4.48	4.28				

OBSERVATIONS.

The numbers given represent the oscillations in an hour of mean time of a *simple* pendulum, measuring from the point of suspension to the centre of a heavy spherical bob attached to a fine thread and oscillating through an exceedingly small arc in a vacuum.

The compound or material pendulum employed for regulating horological trains will give the number of oscillations indicated in the table when the length set opposite that number is equal to the distance between the centres of suspension and oscillation (see article **1282**).

The assumption that the centre of oscillation coincides approximately with the point at which the pendulum will rest horizontally on a knife-edge (**1016**) is only legitimate when the rod is very light and the weight of the pendulum acts nearly through the centre of the bob.

The watchmaker will do well to employ a small platinum ball suspended in front of a carefully graduated vertical rule by a fine thread that can be lengthened or shortened at will. If the point of suspension is determined by a clamp that is opened or closed by a set screw, it will be easy to adjust this pendulum to the length indicated in the table, and, by making it oscillate side by side with the

compound pendulum under consideration, to ascertain the approximate position of the centre of suspension of this latter.

The length of the pendulum giving 1 oscillation in an hour (12,880,337·93 metres or 507,109,080 inches) affords a useful datum for certain calculations with reference to the lengths of pendulum (see article **1277**). For an oscillation of 2° the lower end will travel through a space of 280 miles.

In the above table all dimensions are given in metres and millimetres. If it is required to express them in feet and inches the necessary conversions can be at once effected in any given case by employing the conversion table below.

TABLES FOR THE MUTUAL CONVERSION OF FRENCH AND ENGLISH MEASURES OF LENGTH AND WEIGHT.

Length.

Inches expressed in *Millimetres and French Lines.*			*Millimetres* expressed in *Inches and French Lines.*			*French Lines* expressed in *Inches and Millimetres.*		
Inches.	Equal to		Millimetres.	Equal to		French Lines.	Equal to	
	Millimetres.	French Lines		Inches.	French Lines.		Inches.	Millimetres.
1	25.39954	11.25951	1	0.0393708	0.44329	1	0.088814	2.25583
2	50.79908	22.51903	2	0.0787416	0.88659	2	0.177628	4.51166
3	76.19862	33.77854	3	0.1181124	1.32989	3	0.266441	6.76749
4	101.59816	45.03806	4	0.1574832	1.77318	4	0.355255	9.02332
5	126.99771	56.29757	5	0.1968539	2.21648	5	0.444069	11.27915
6	152.39725	67.55709	6	0.2362247	2.65978	6	0.532883	13.53497
7	177.79679	78.81660	7	0.2755955	3.10307	7	0.621697	15.79080
8	203.19633	90.07612	8	0.3149664	3.54637	8	0.710510	18.04663
9	228.59587	101.33563	9	0.3543371	3.98966	9	0.799324	20.30246
10	253.99541	112.59515	10	0.3937079	4.43296	10	0.888138	22.55829
						11	0.976952	24.81412
						12	1.065766	27.06995

A metre is the forty millionth part of a meridian of the earth.

1 metre,
10 decimetres,
100 centimetres, or
1000 millimetres
} is equivalent to {
39.37079 inches,
3.28090 feet,
1.09363 yards, or
0.00062138 miles.

1 sq. centimetre = 0.15501 sq. inch. 1 sq. inch = 6.45137 sq. centimetres.
1 cub. „ = 0.06103 cub. „ 1 cub. „ = 16.38618 cub. „

TABLES FOR THE MUTUAL CONVERSION OF FRENCH AND ENGLISH MEASURES OF LENGTH AND WEIGHT—*continued.*

WEIGHT.

Troy Grains expressed in Grammes		Troy Ounces expressed in Grammes.		Avoirdupois ounces expressed in Grammes.		Grammes expressed in Troy Grains and Ounces and Avoirdupois Ounces.			
							Equal to		
Troy Grains.	Equal to Grammes.	Troy Ounces.	Equal to Grammes.	Avoir. Ounces.	Equal to Grammes.	Grammes.	Troy Grains.	Troy Ounces.	Avoir. Ounces.
1	0.06479895	1	31.10350	1	28.34954	1	15.43235	0.03215	0.03527
2	0.12959790	2	62.20699	2	56.69908	2	30.86470	0.06430	0.07055
3	0.19439685	3	93.31049	3	85.04862	3	46.29705	0.09645	0.10582
4	0.25919580	4	124.41398	4	113.39816	4	61.72940	0.12860	0.14109
5	0.32399475	5	155.51748	5	141.74770	5	77.16174	0.16075	0.17637
6	0.38879370	6	186.62098	6	170.09724	6	92.59409	0.19290	0.21164
7	0.45359265	7	217.72447	7	198.44679	7	108.02644	0.22505	0.24692
8	0.51839160	8	248.82797	8	226.79633	8	123.45879	0.25721	0.28219
9	0.58319055	9	279.93147	9	255.14587	9	138.89114	0.28936	0.31747
10	0.64798950	10	311.03496	10	283.49540	10	154.32349	0.32151	0.35274

1 Kilogramme,
10 Hectogrammes,
100 Dekagrammes,
1,000 Grammes
10,000 Decigrammes,
100,000 Centigrammes, or
1,000,000 Milligrammes

} is equivalent to { 2.204621 Avoir. lbs.
2.679227 Troy lbs.

The kilogramme is the weight of 1 cubic decimetre of pure water at its point of maximum density, 4° C (39° F). This volume is called a *litre*, and is equal to 61·02705 cubic inches.

To convert degrees centigrade into degrees Fahrenheit multiply by 9, divide by 5, and add 32.

To convert degrees Fahrenheit into degrees centigrade subtract 32, multiply the remainder by 5, and divide by 9.

To convert degrees Reaumur into degrees Fahrenheit multiply by 9, divide by 4, and add 32.

Numerous experimental data will be found in the tabular summary on pages 52-3.

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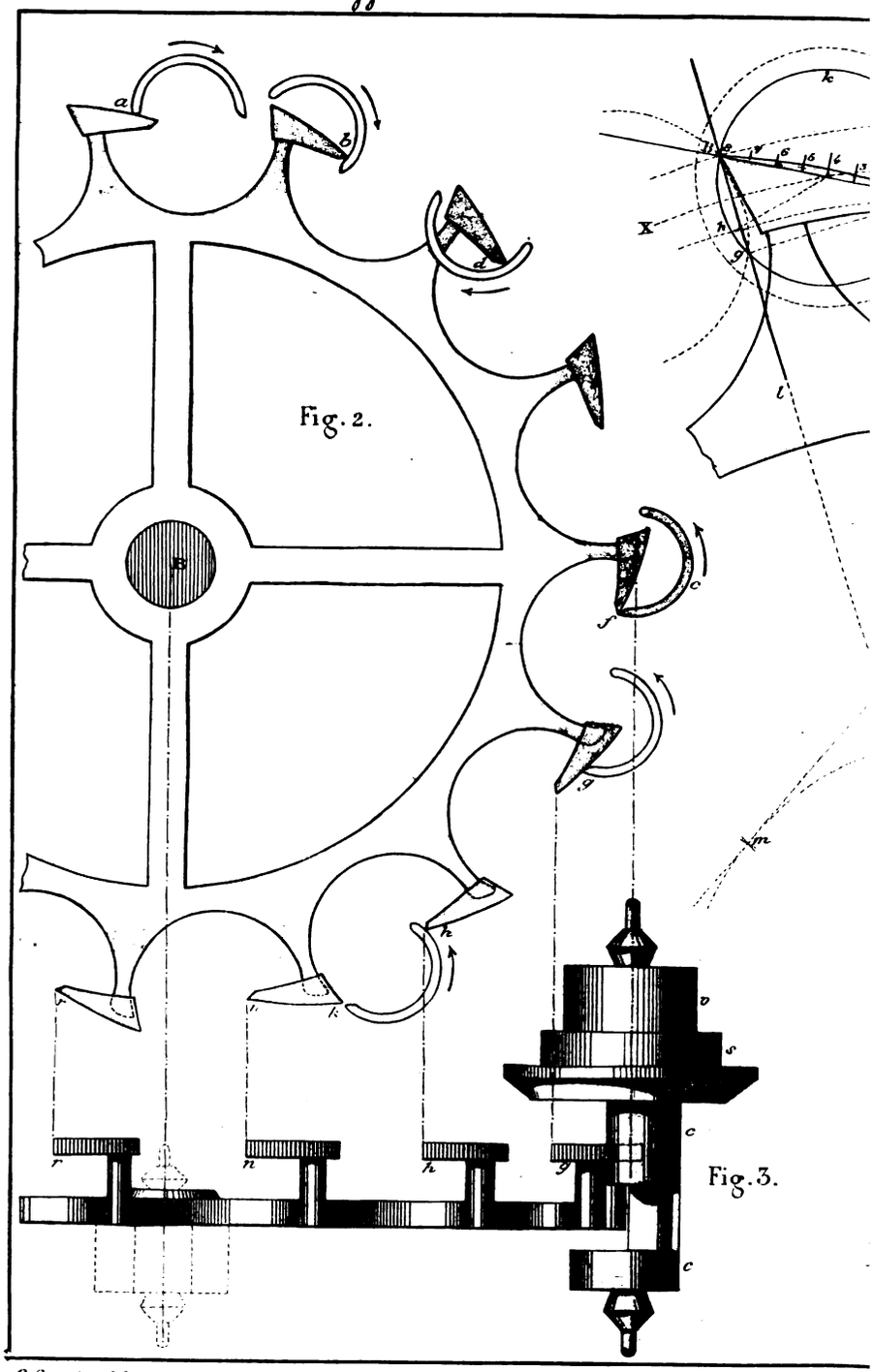

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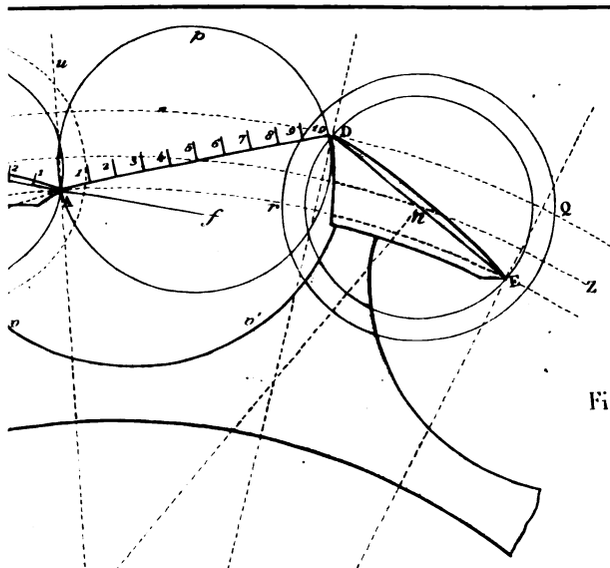

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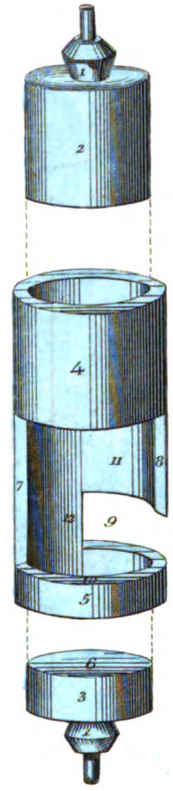

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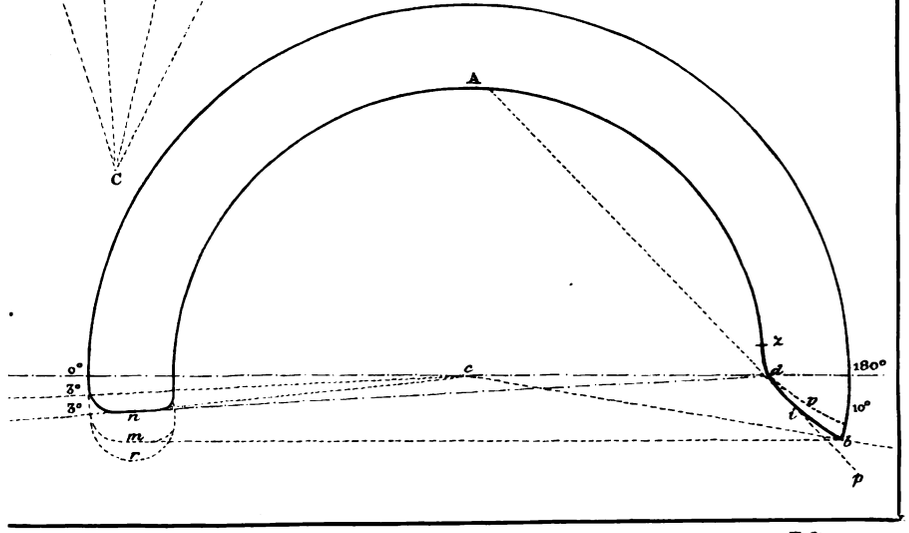

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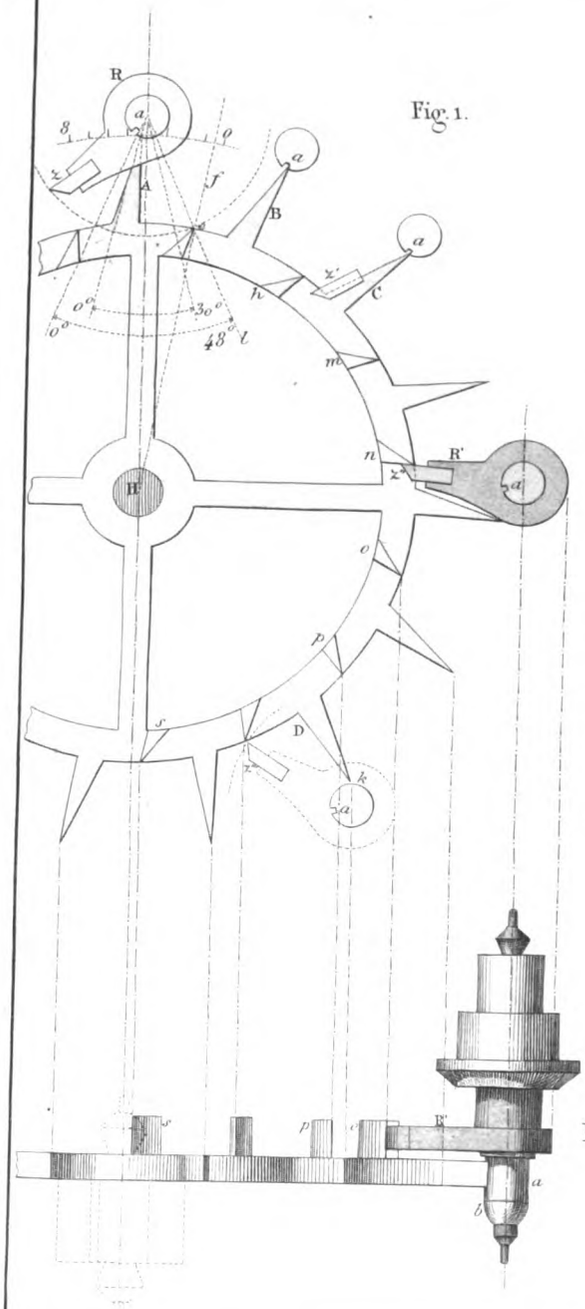

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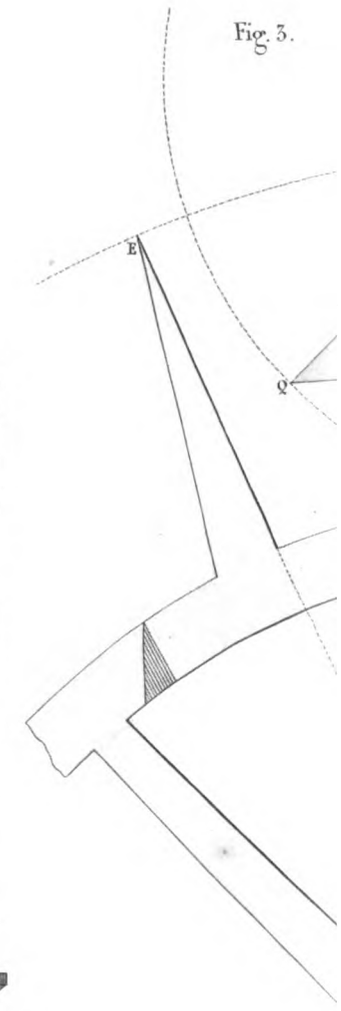

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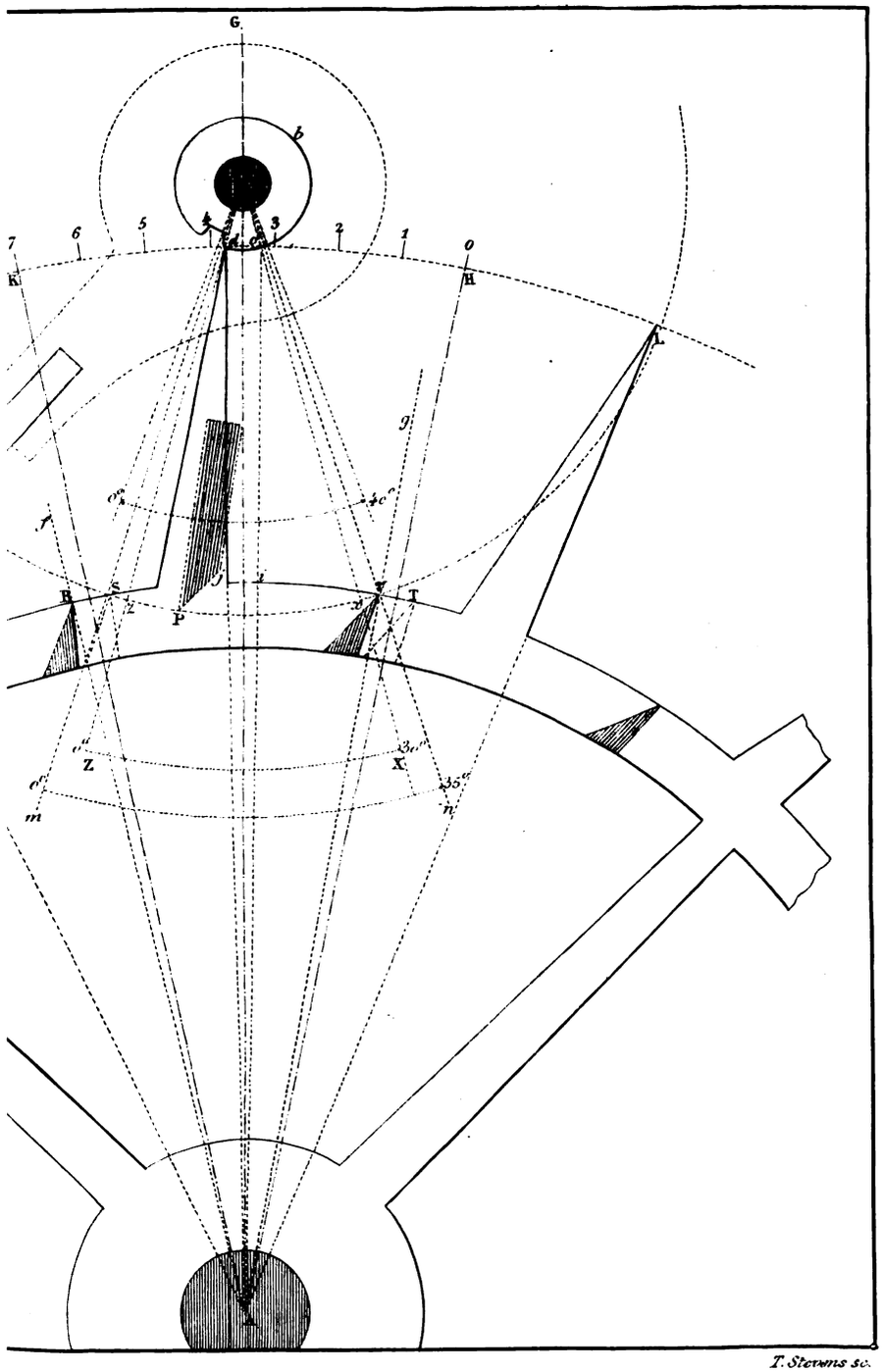

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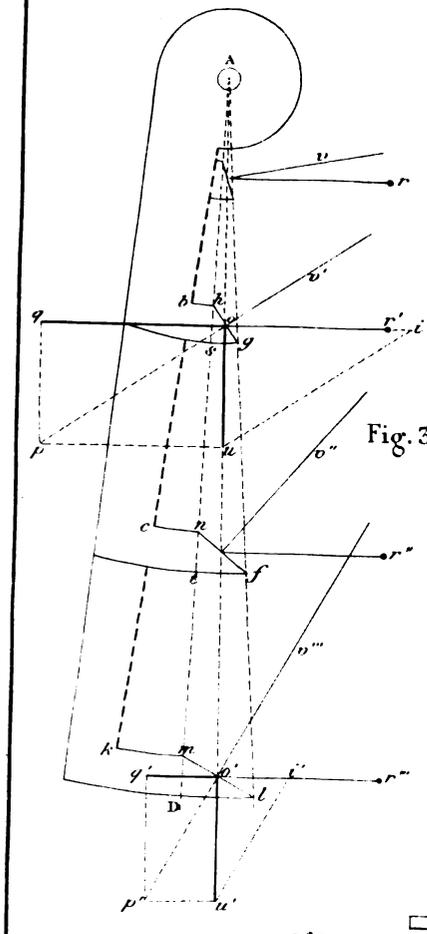

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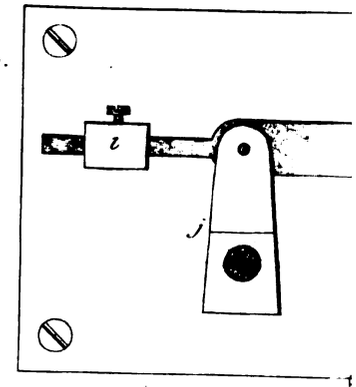

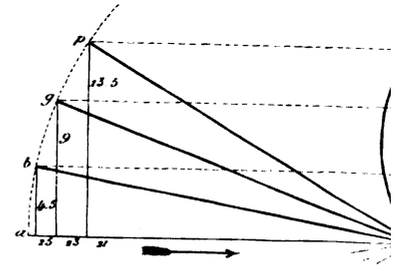

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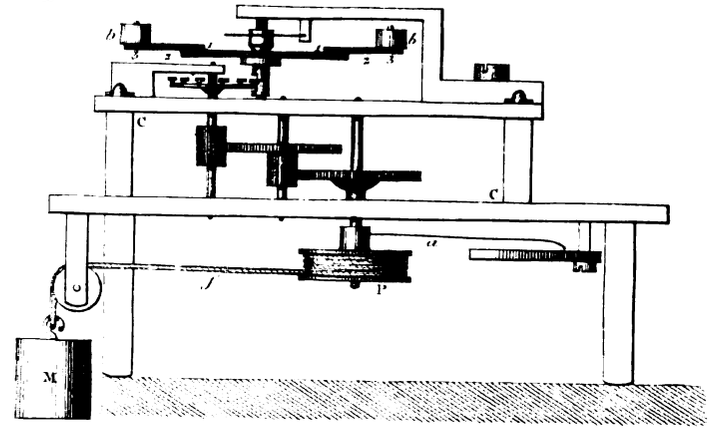

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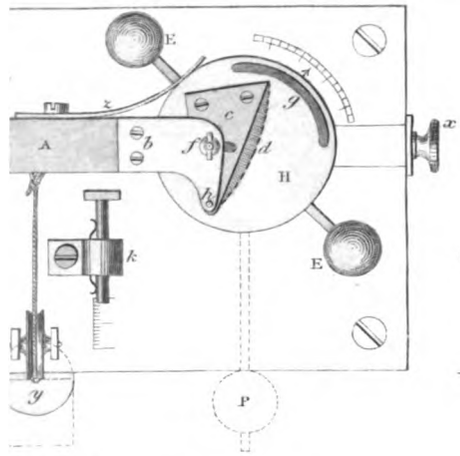

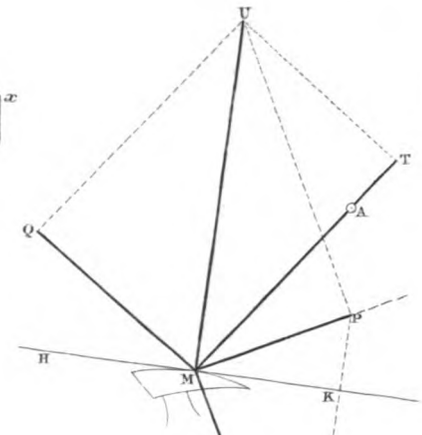

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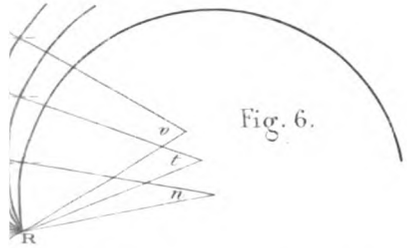

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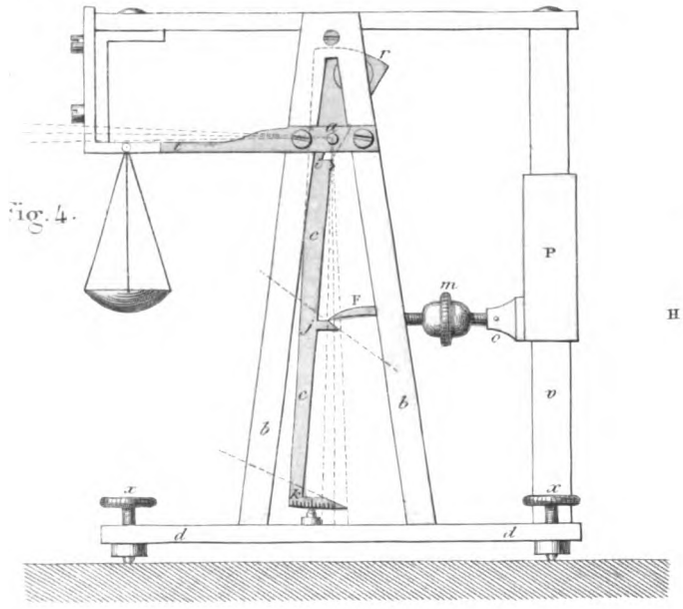

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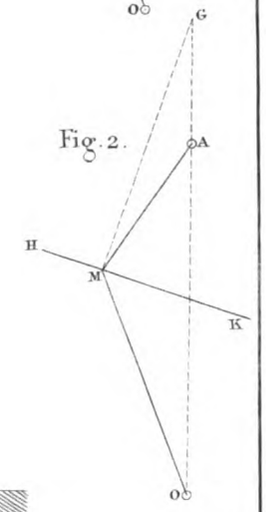

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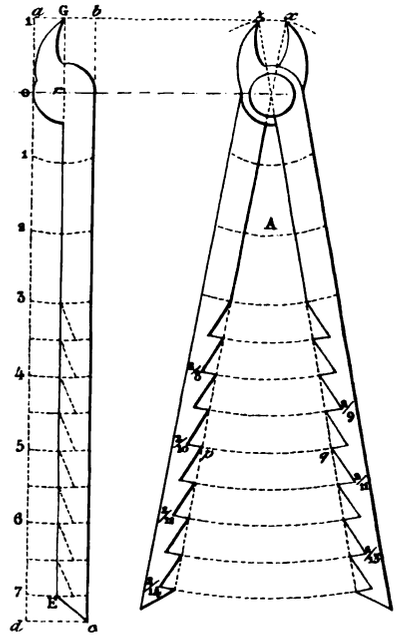

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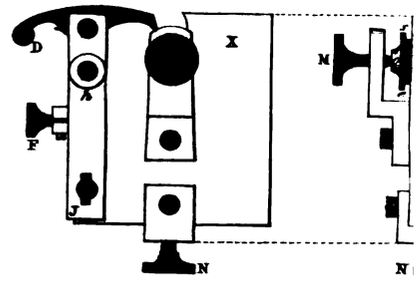

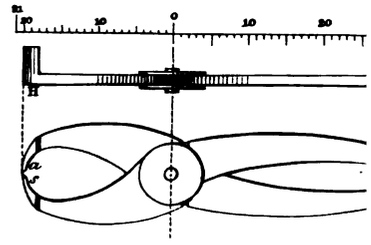

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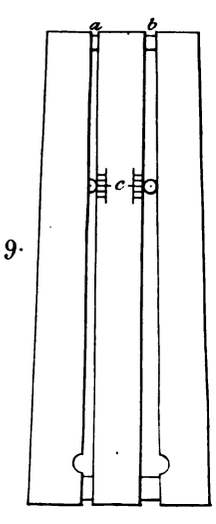

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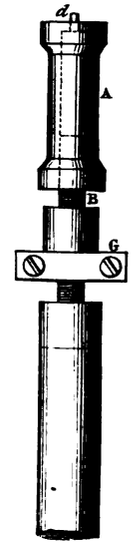

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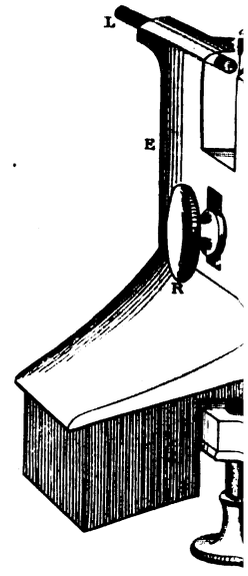

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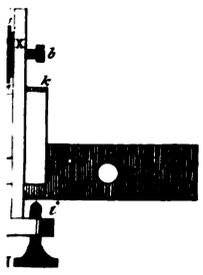

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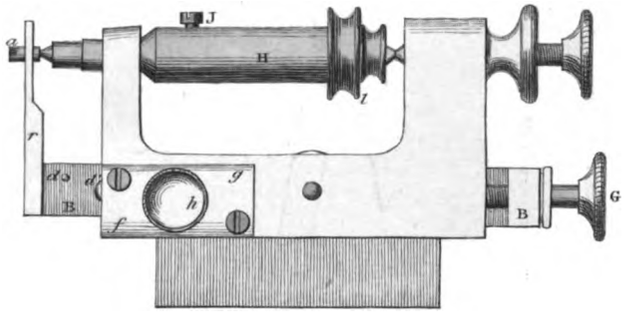
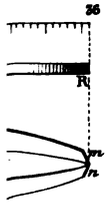




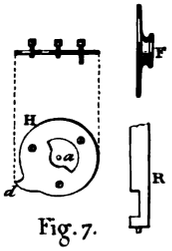
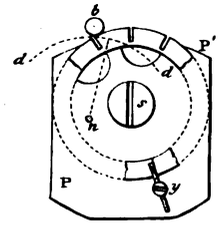


Fig. 7.



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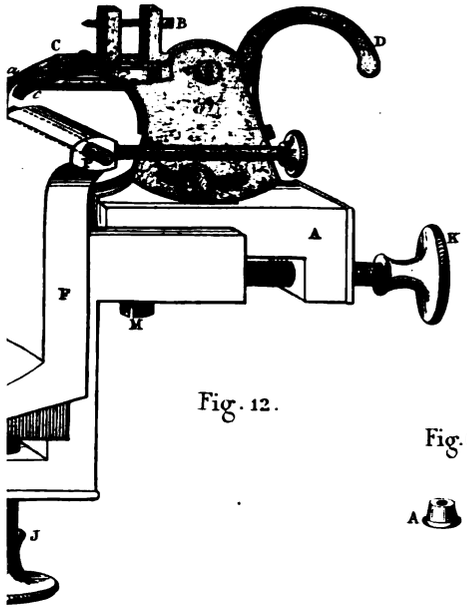
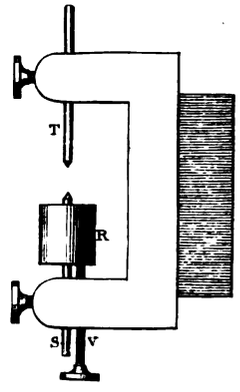


Fig. 12.

Fig. 11.



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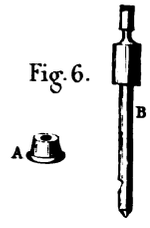

T. Stevens sc.

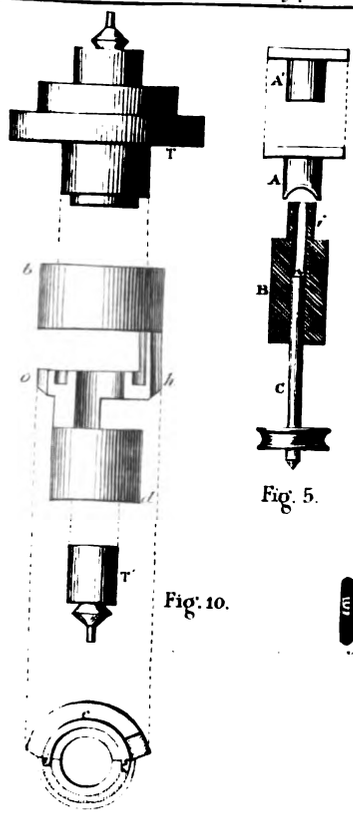

Fig. 10.

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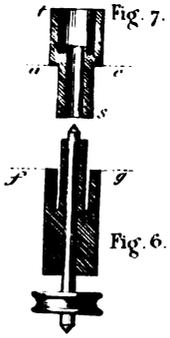

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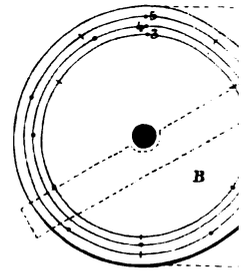

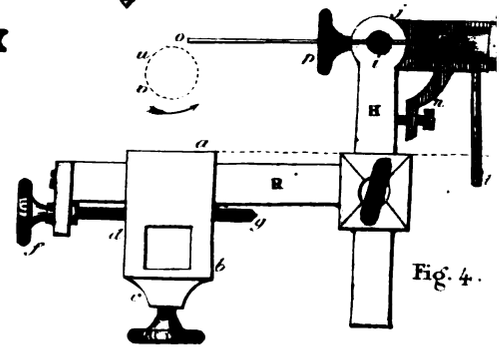

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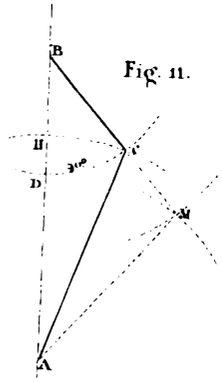

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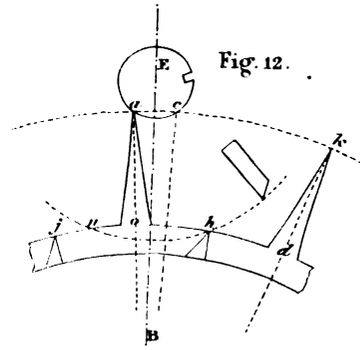

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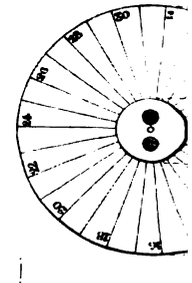
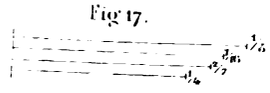
Fig





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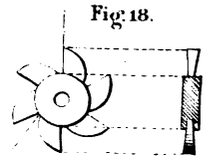

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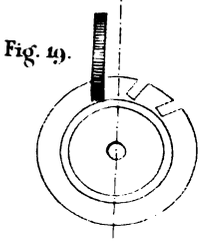

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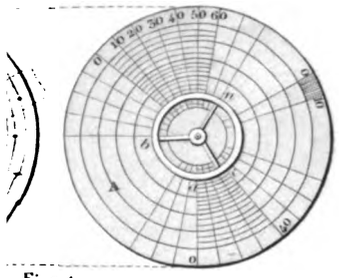

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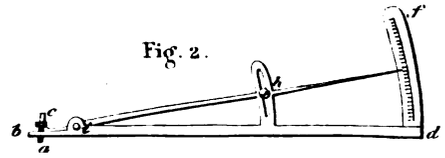

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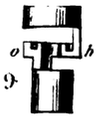

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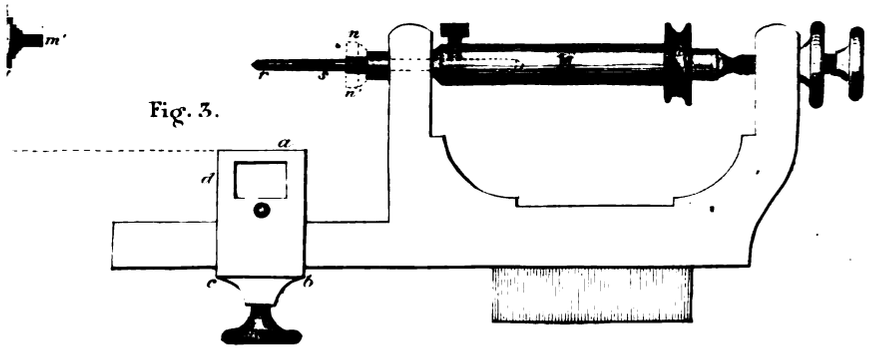

Fig. 3.

Fig. 15.

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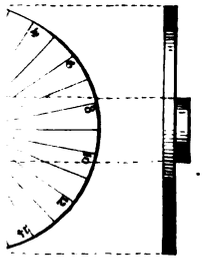

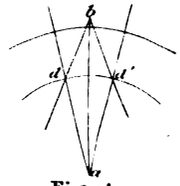

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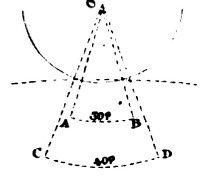

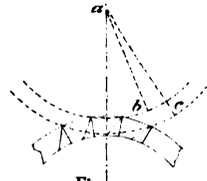

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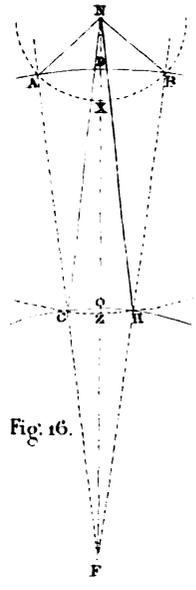

Fig. 16.

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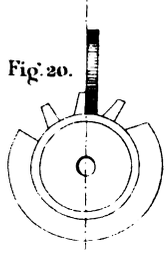

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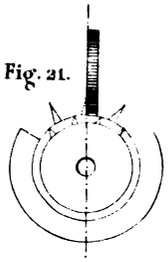

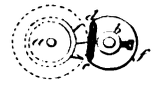

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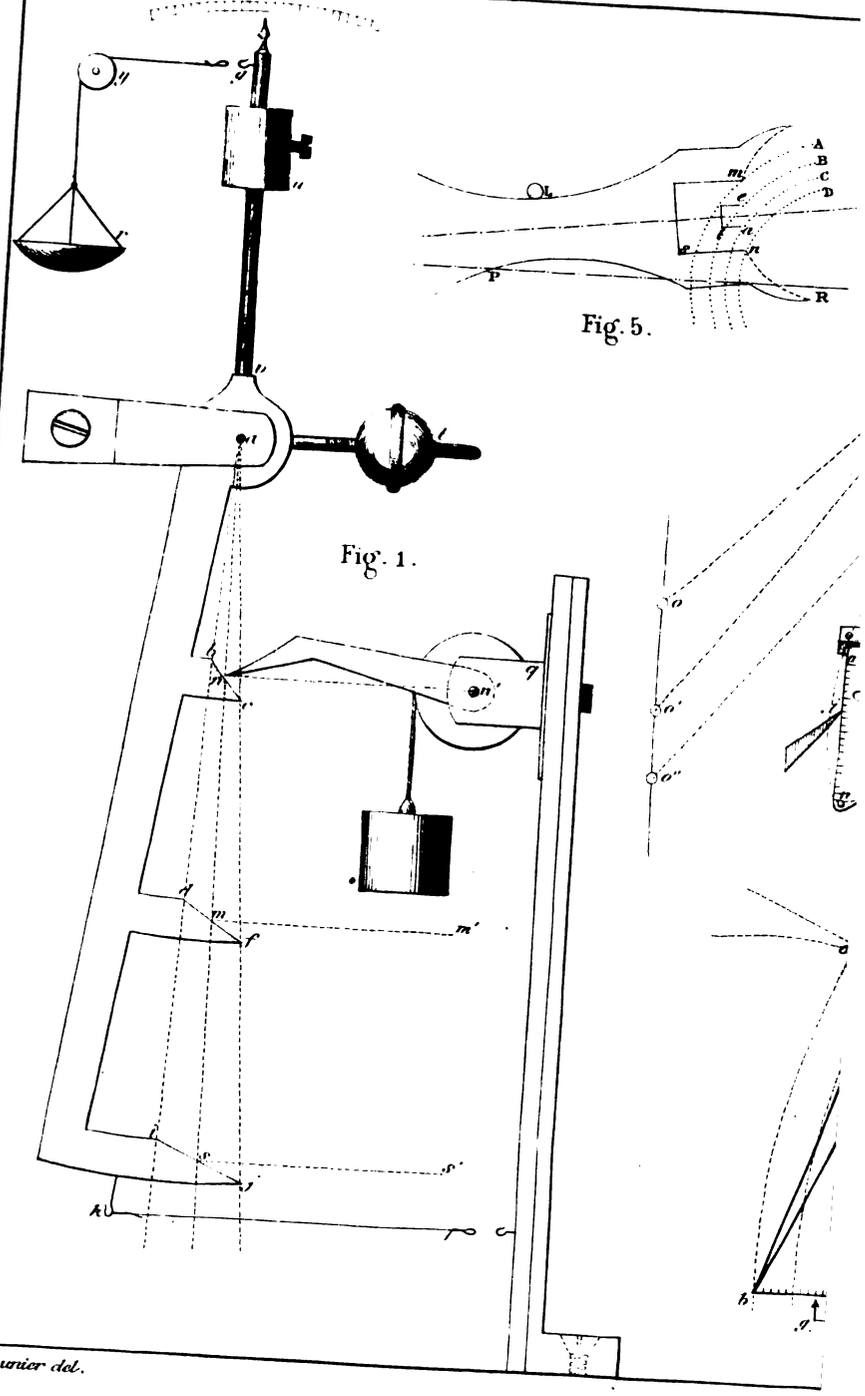
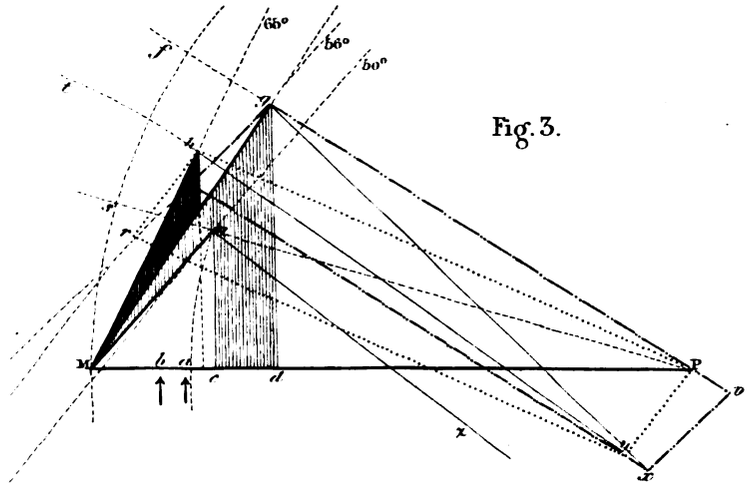


Fig. 1.

Fig. 5.

C. Saunier del.



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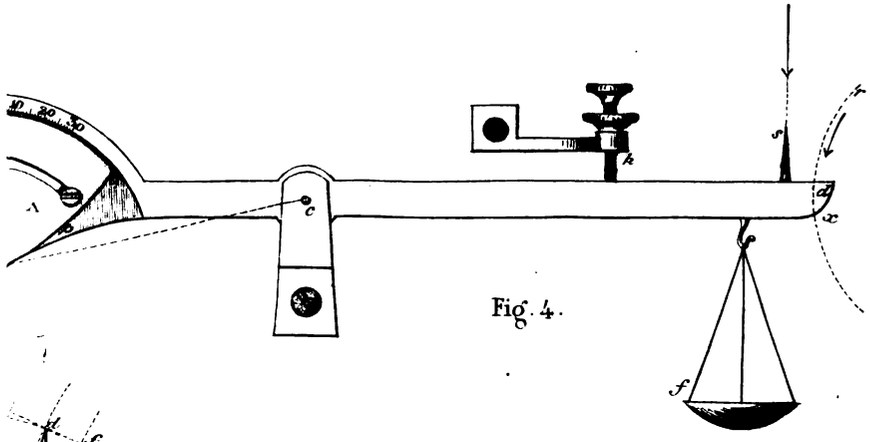

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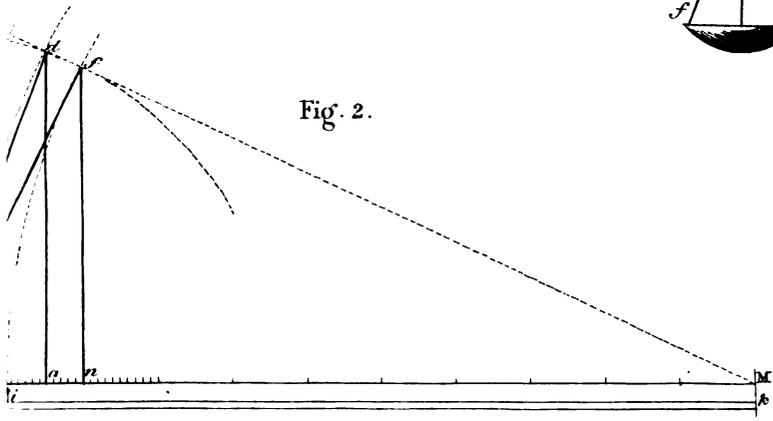

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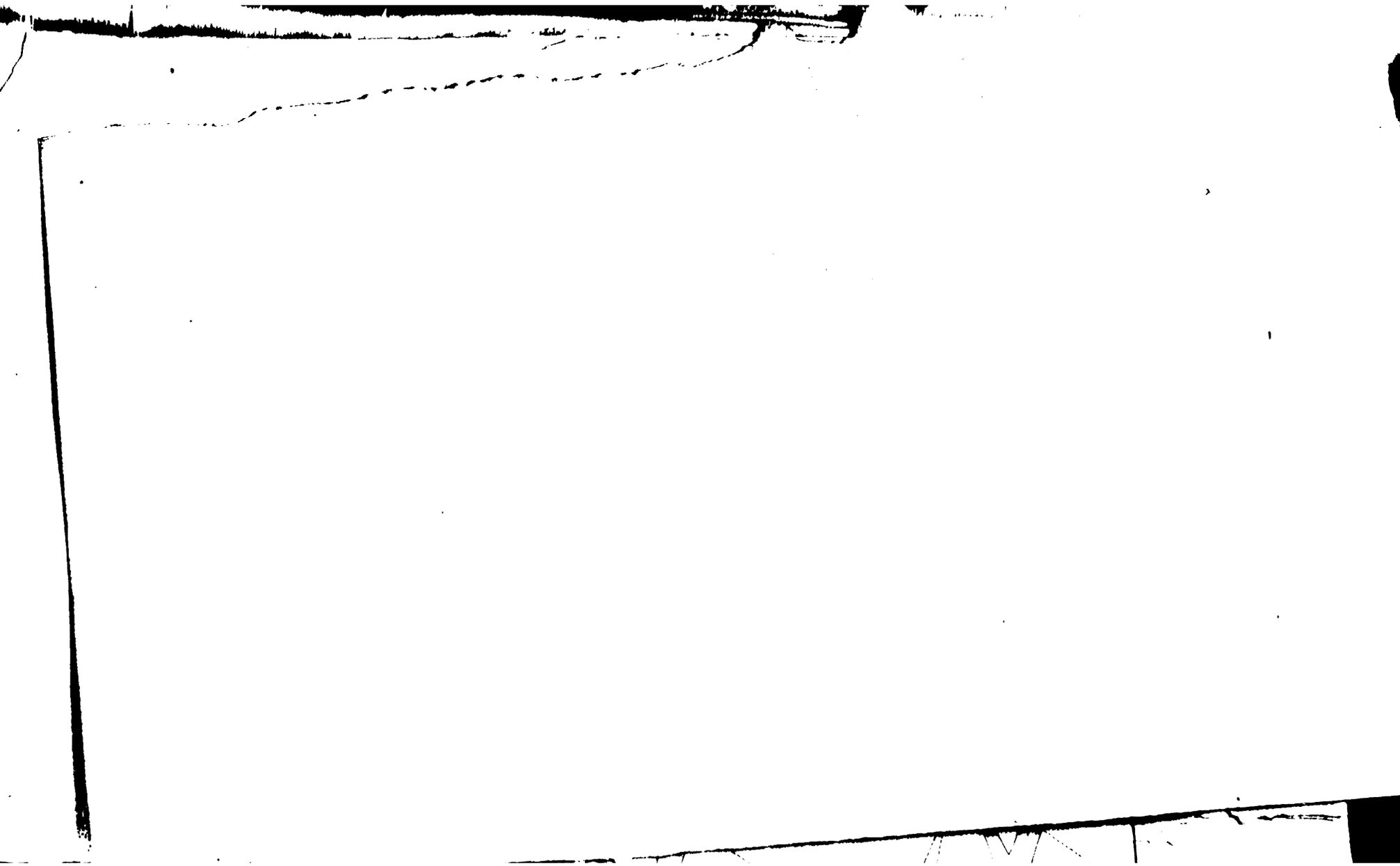
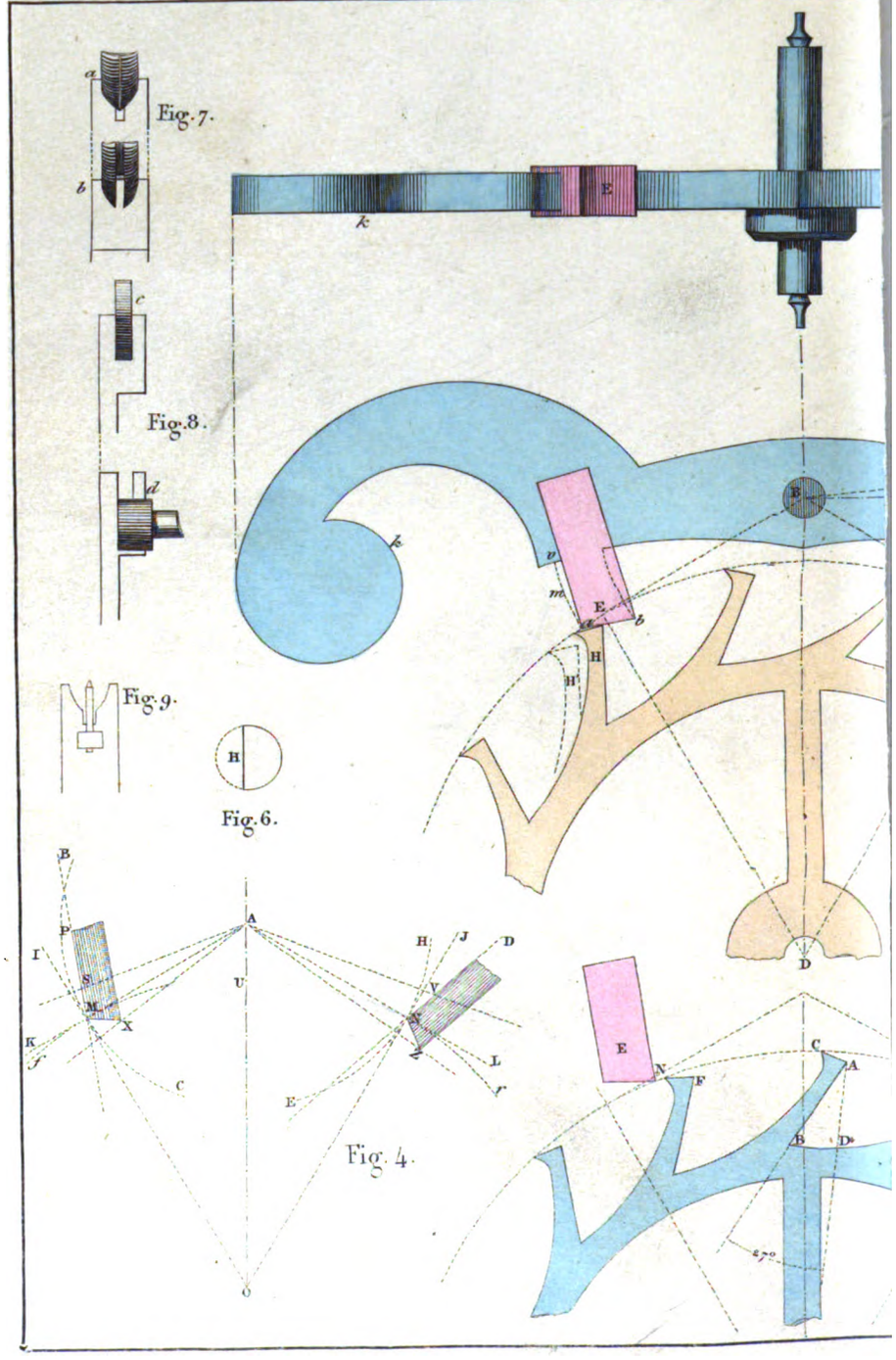




C. Saunier del

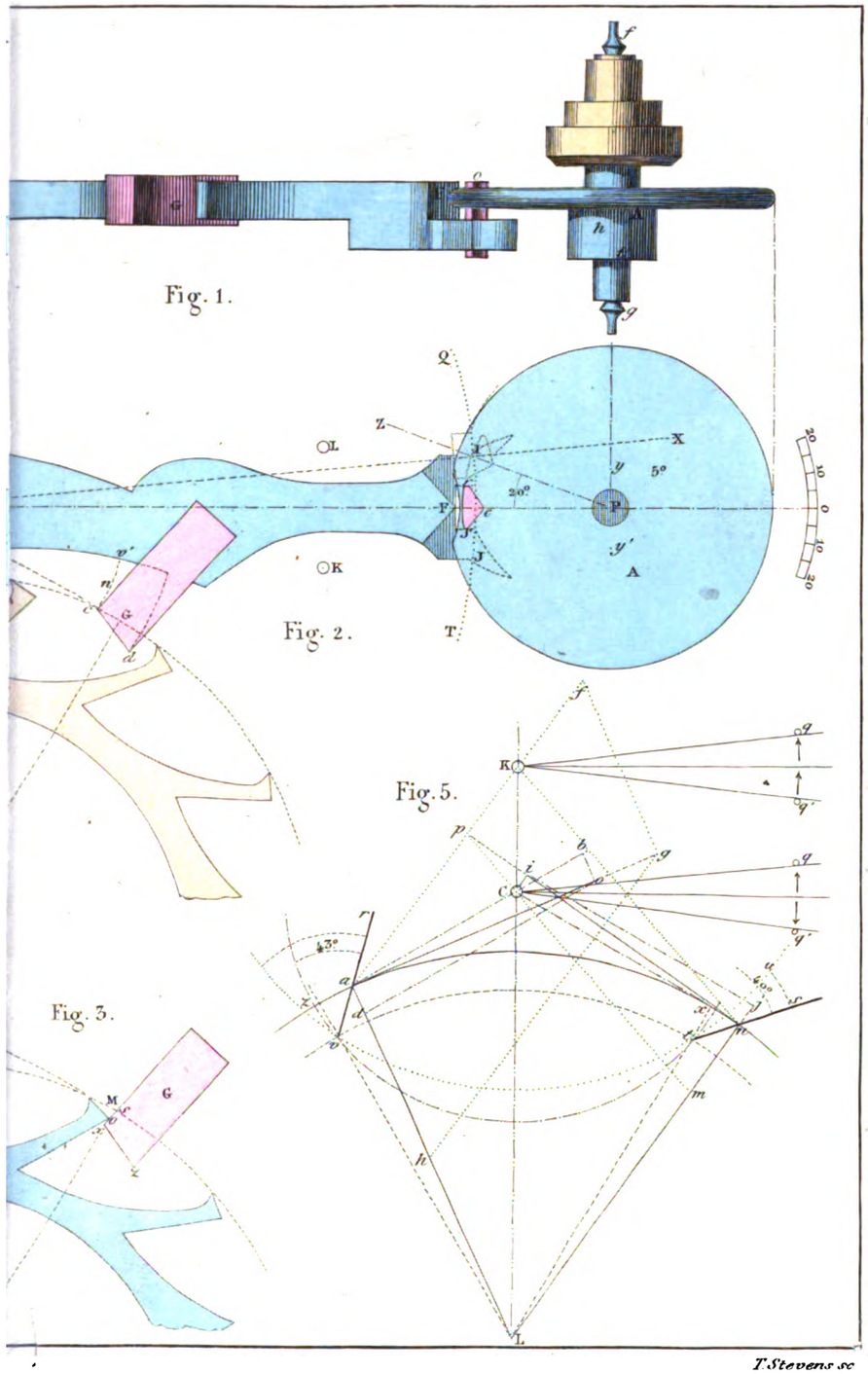
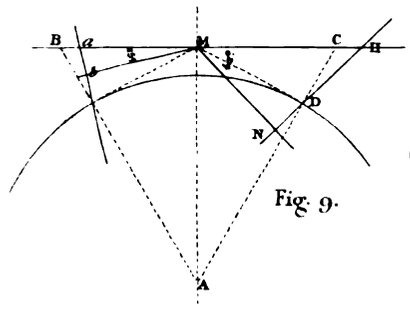


Fig. 1.
Fig. 2.
Fig. 3.
Fig. 5.
T. Stevens sc



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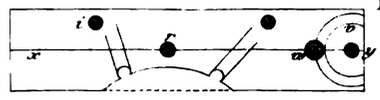

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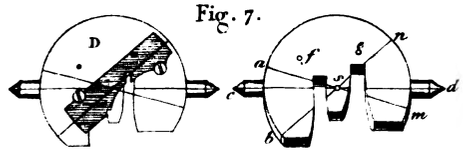

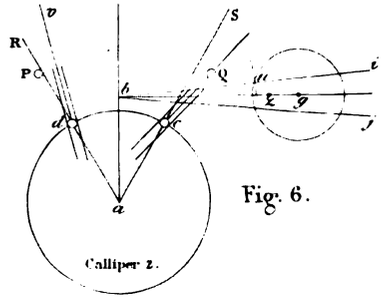

Fig. 6.

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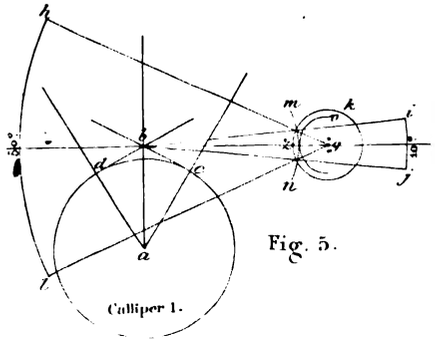

Fig. 5.

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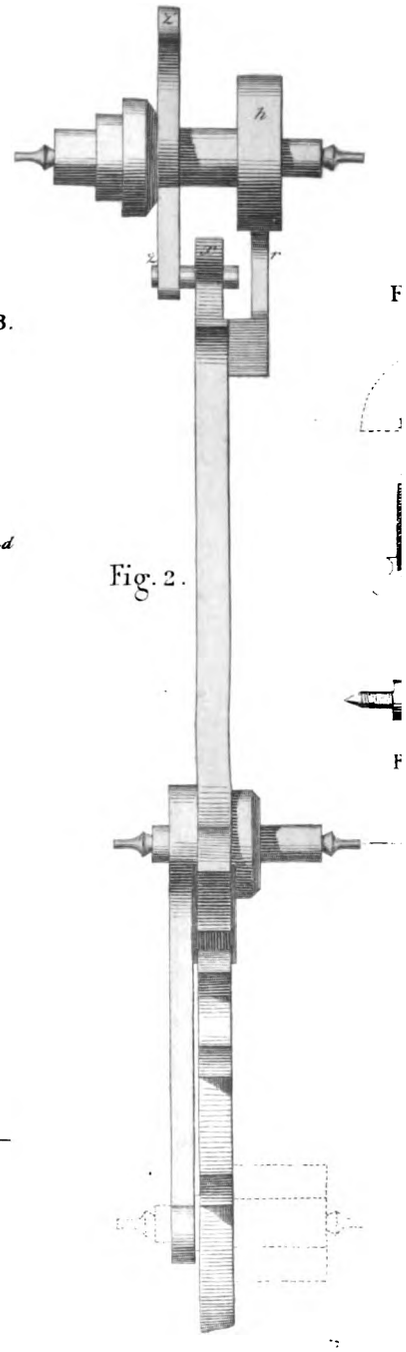

Fig. 2.

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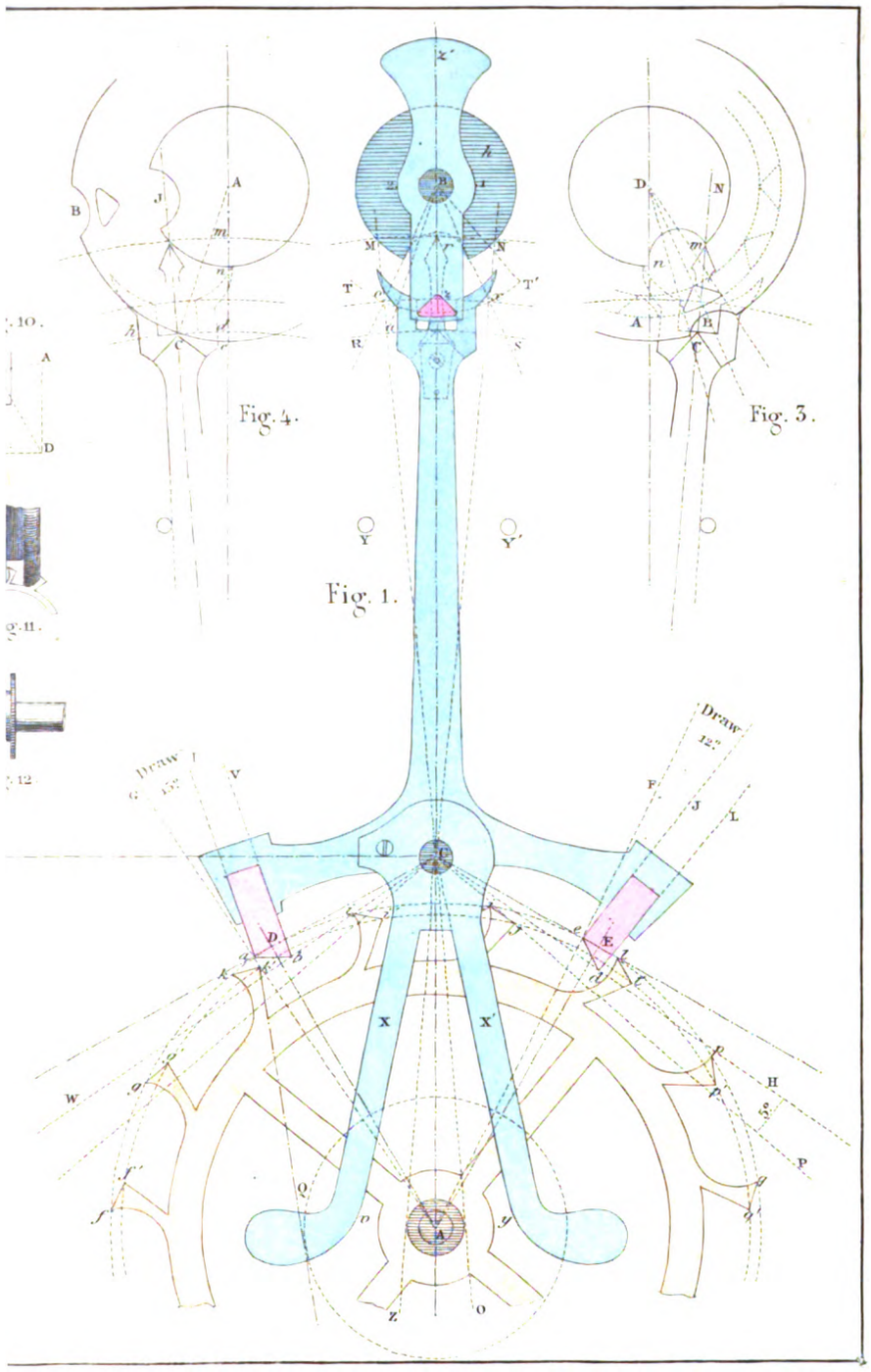
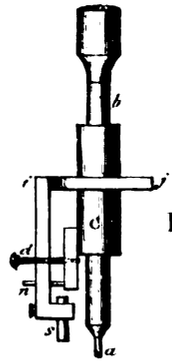


Fig. 4.
Fig. 3.
Fig. 1.
Draw 15°
Draw 12°
T. Stevens sc.



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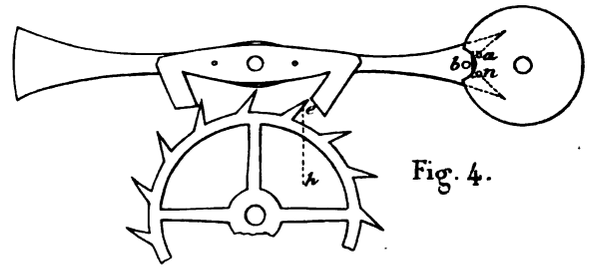

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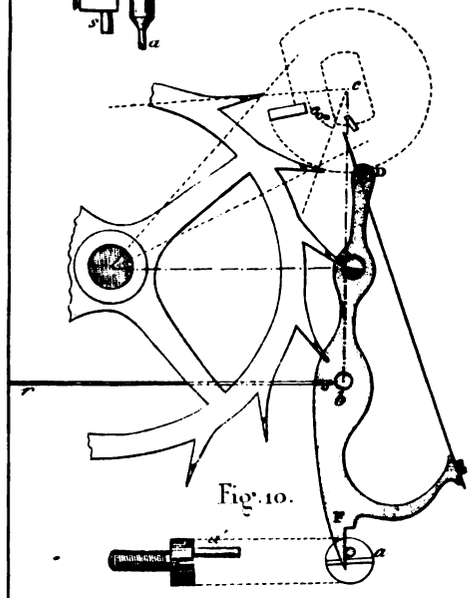

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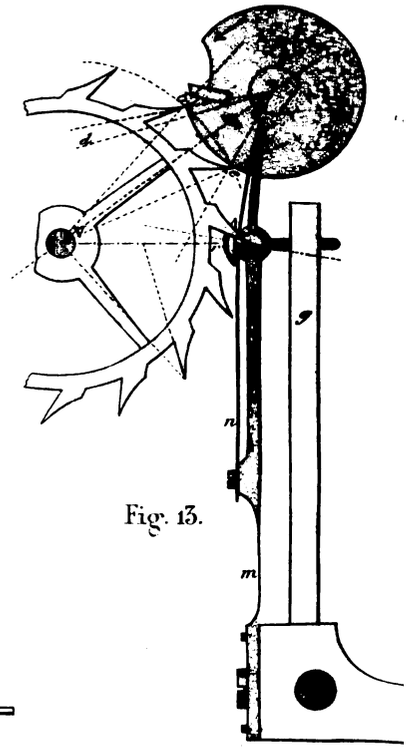

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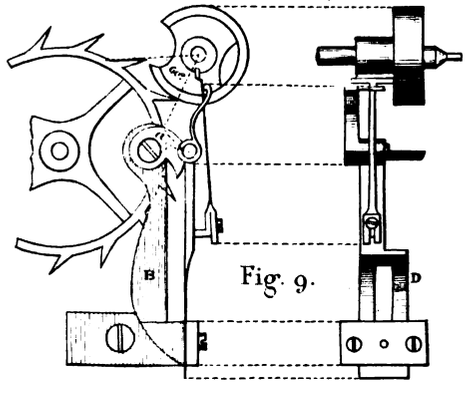

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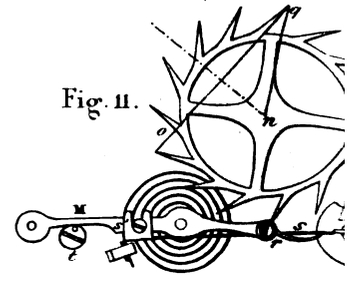
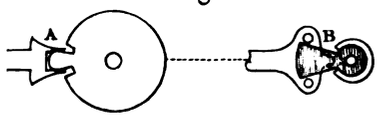


Fig. 11.

C. Saunier del.

Fig. 3.



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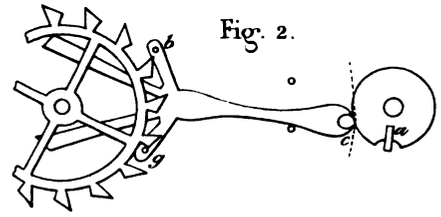

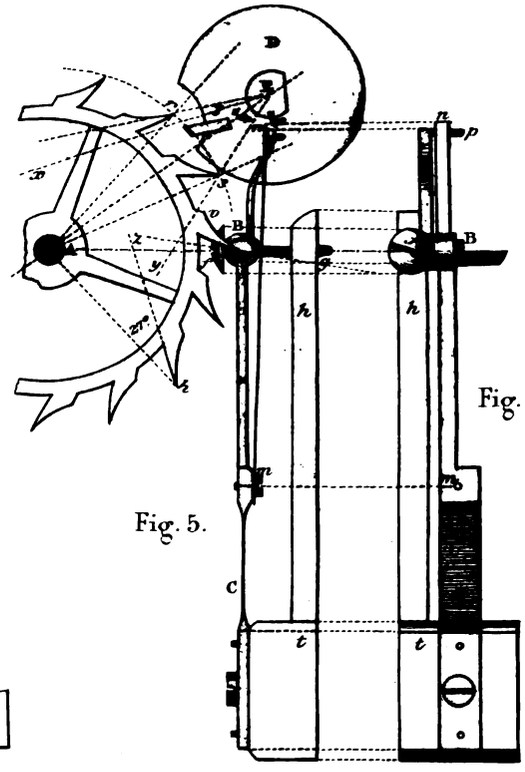

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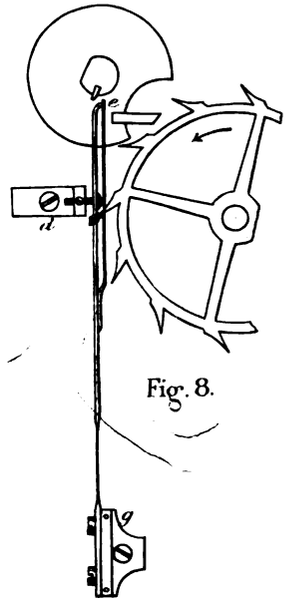

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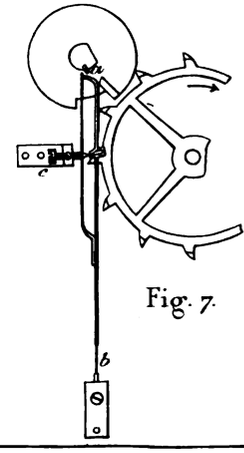

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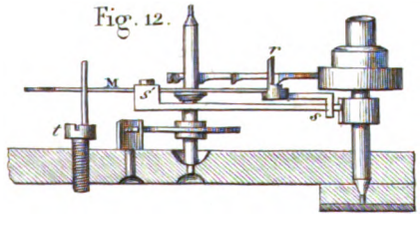


T. Stevens sc

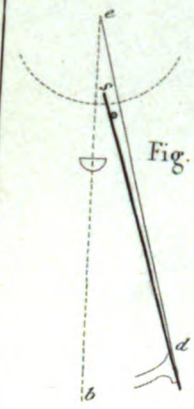
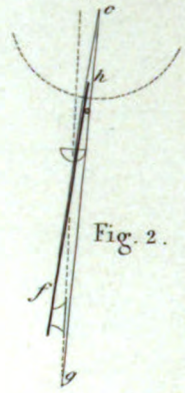


Fig. 1.



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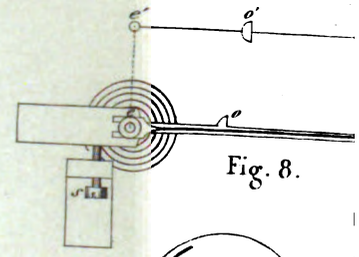

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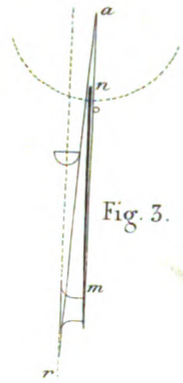

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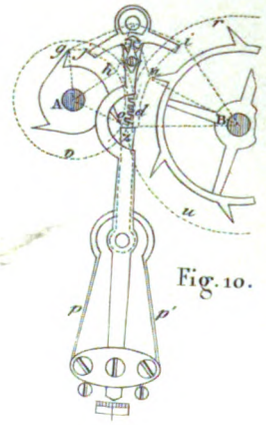

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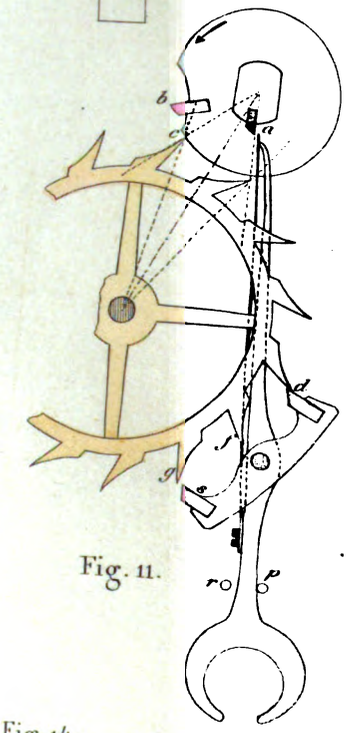

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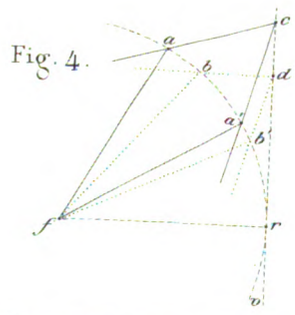

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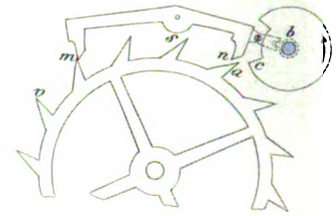

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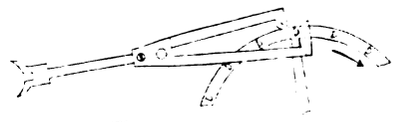

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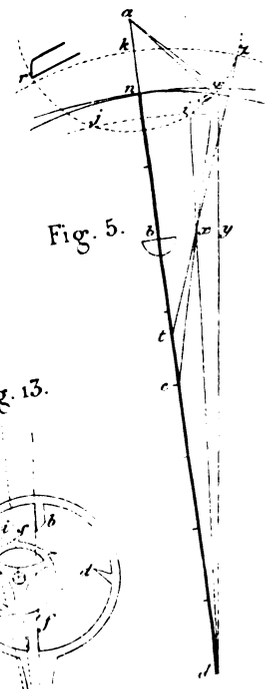

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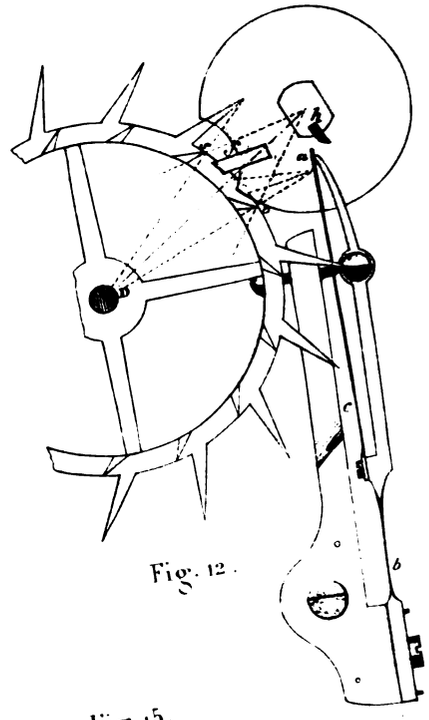

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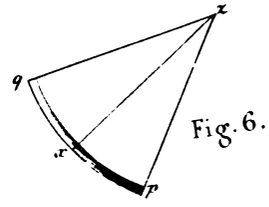

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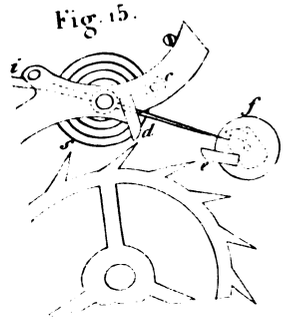

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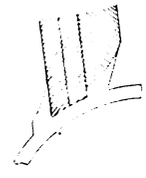

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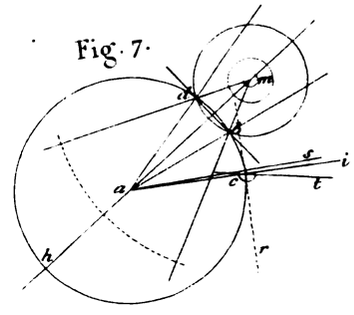

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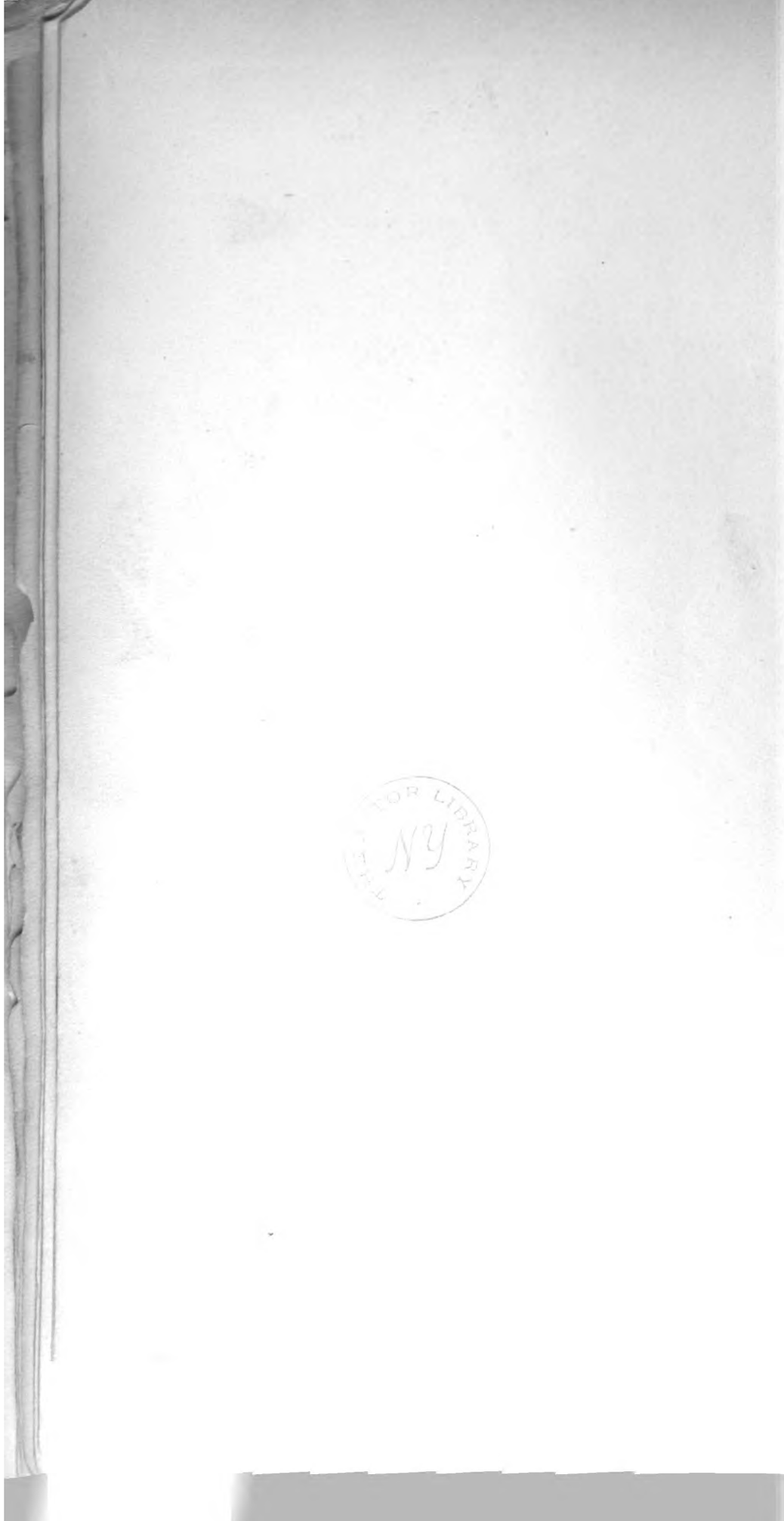

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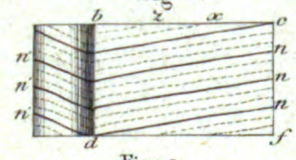

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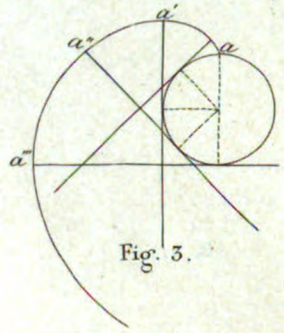

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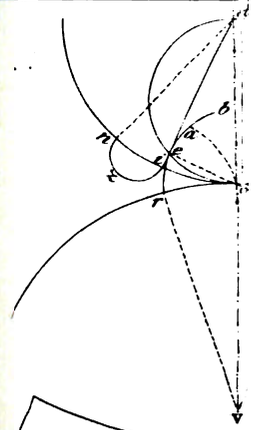

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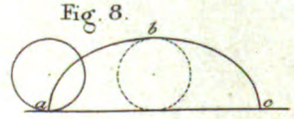

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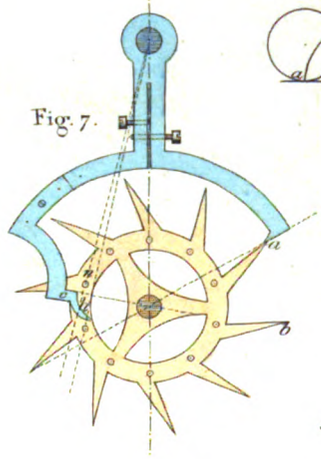

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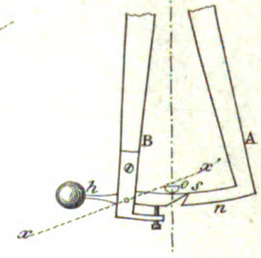

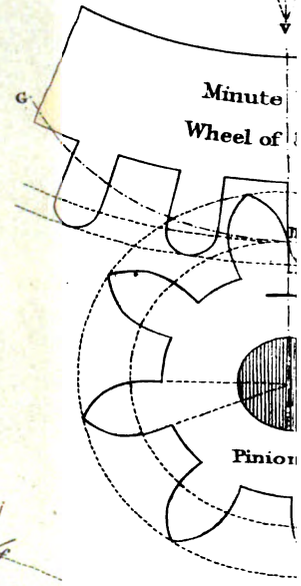

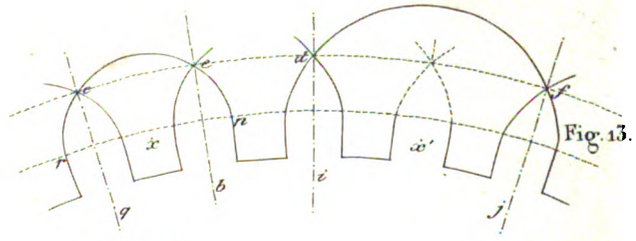

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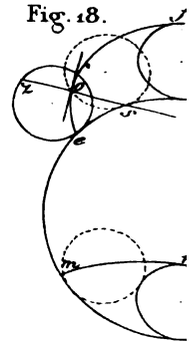

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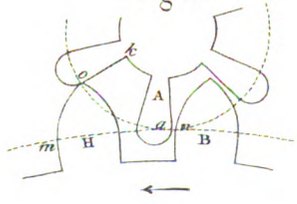

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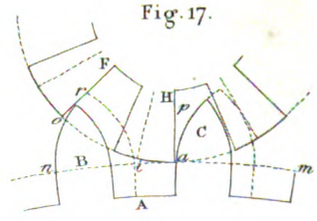

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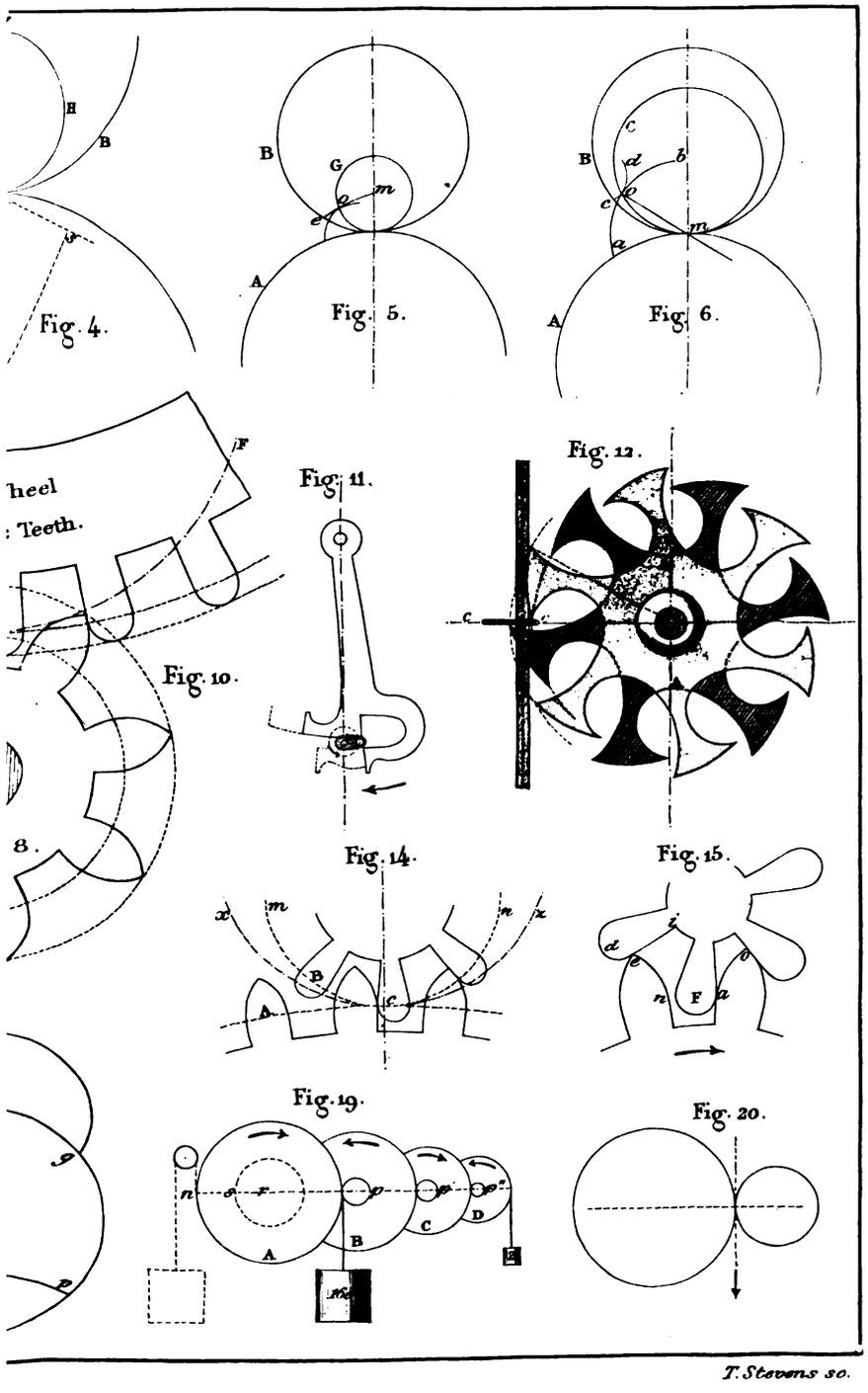
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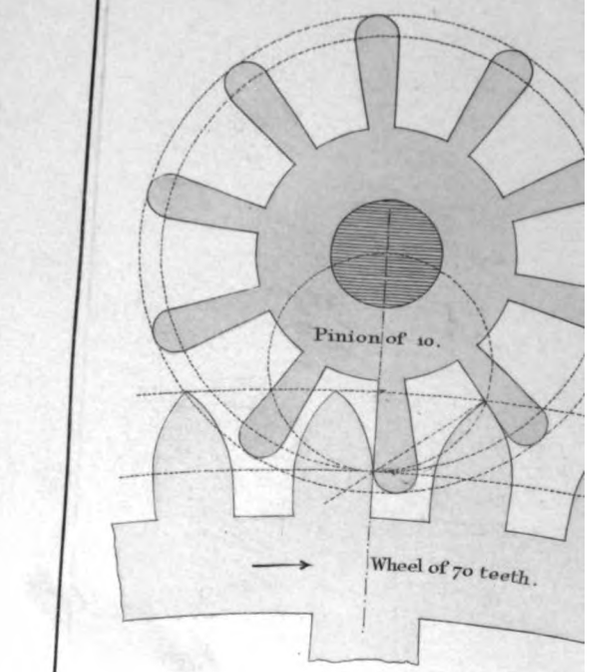

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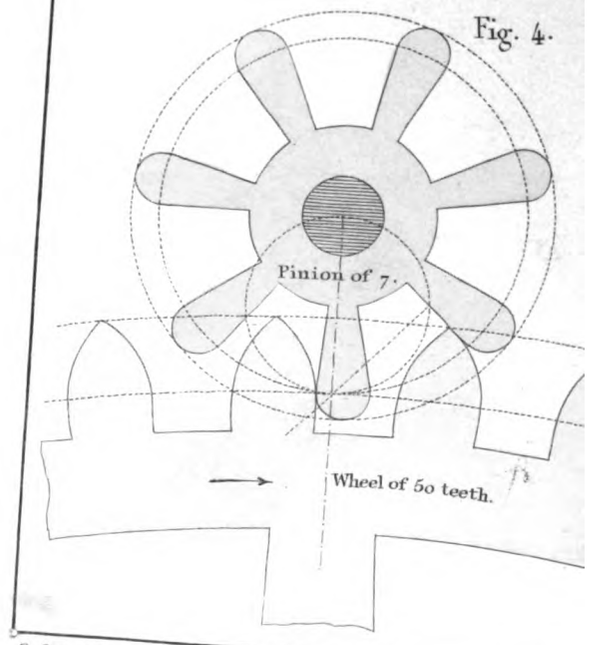

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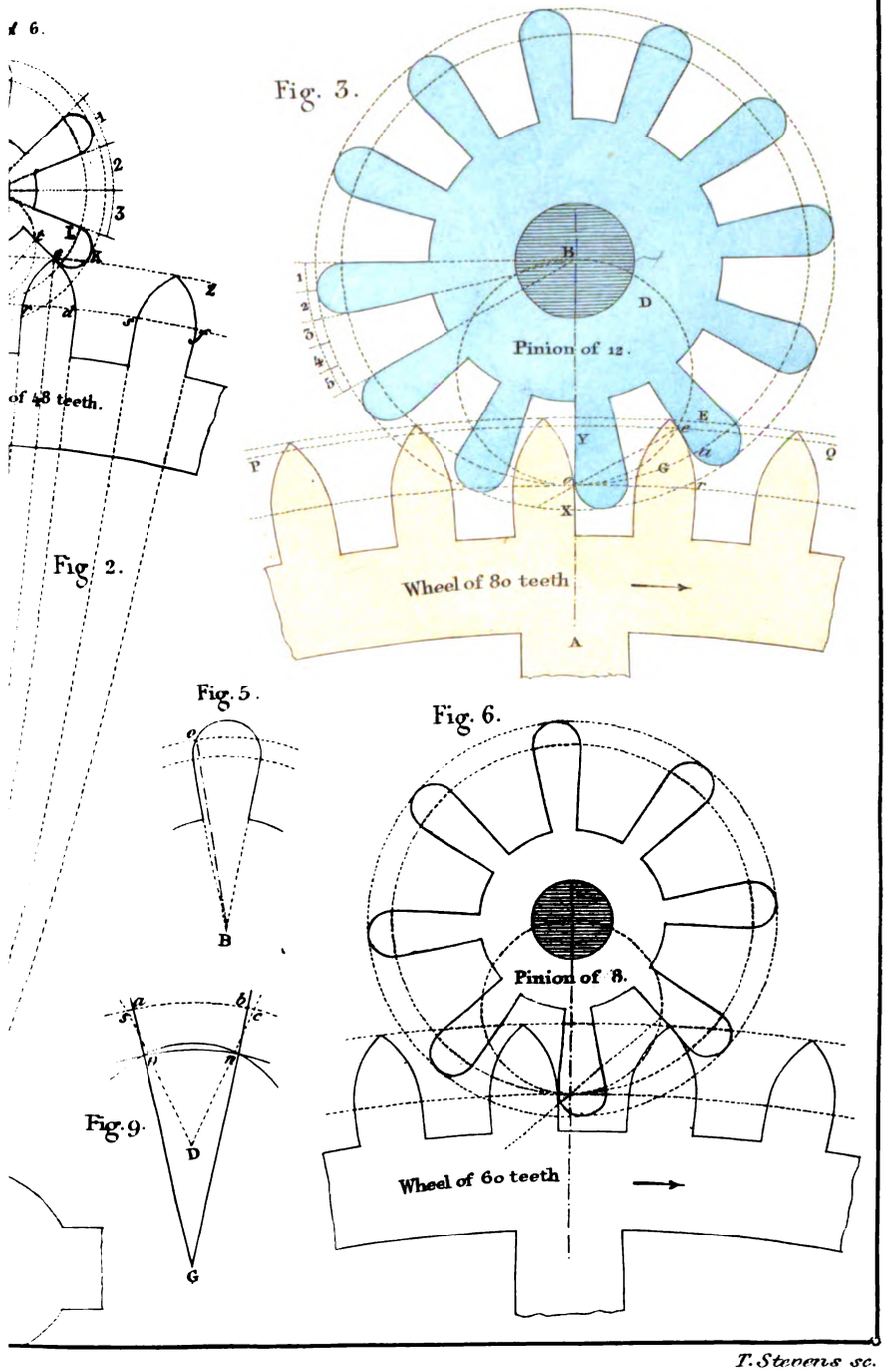
Fig. 3.
Pinion of 12.
Wheel of 80 teeth
Fig. 2.
of 48 teeth.
Fig. 5.
Fig. 6.
Pinion of 8.
Wheel of 60 teeth
Fig. 9.

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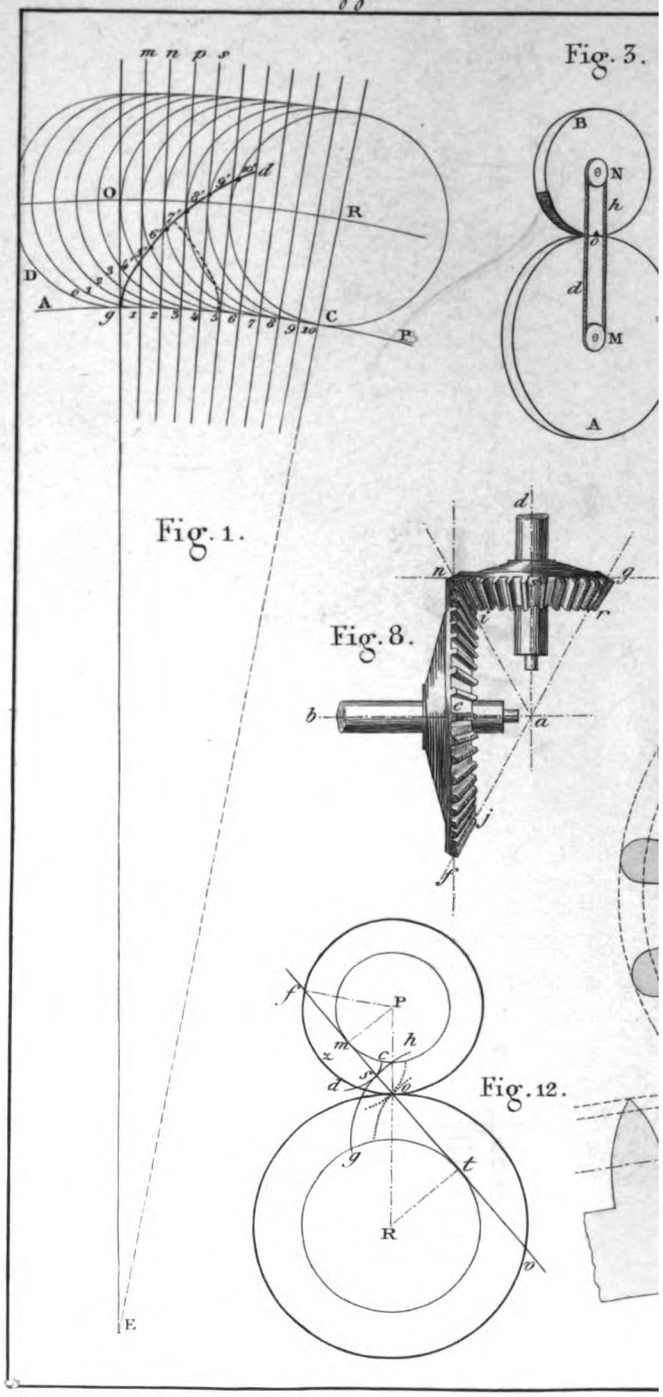
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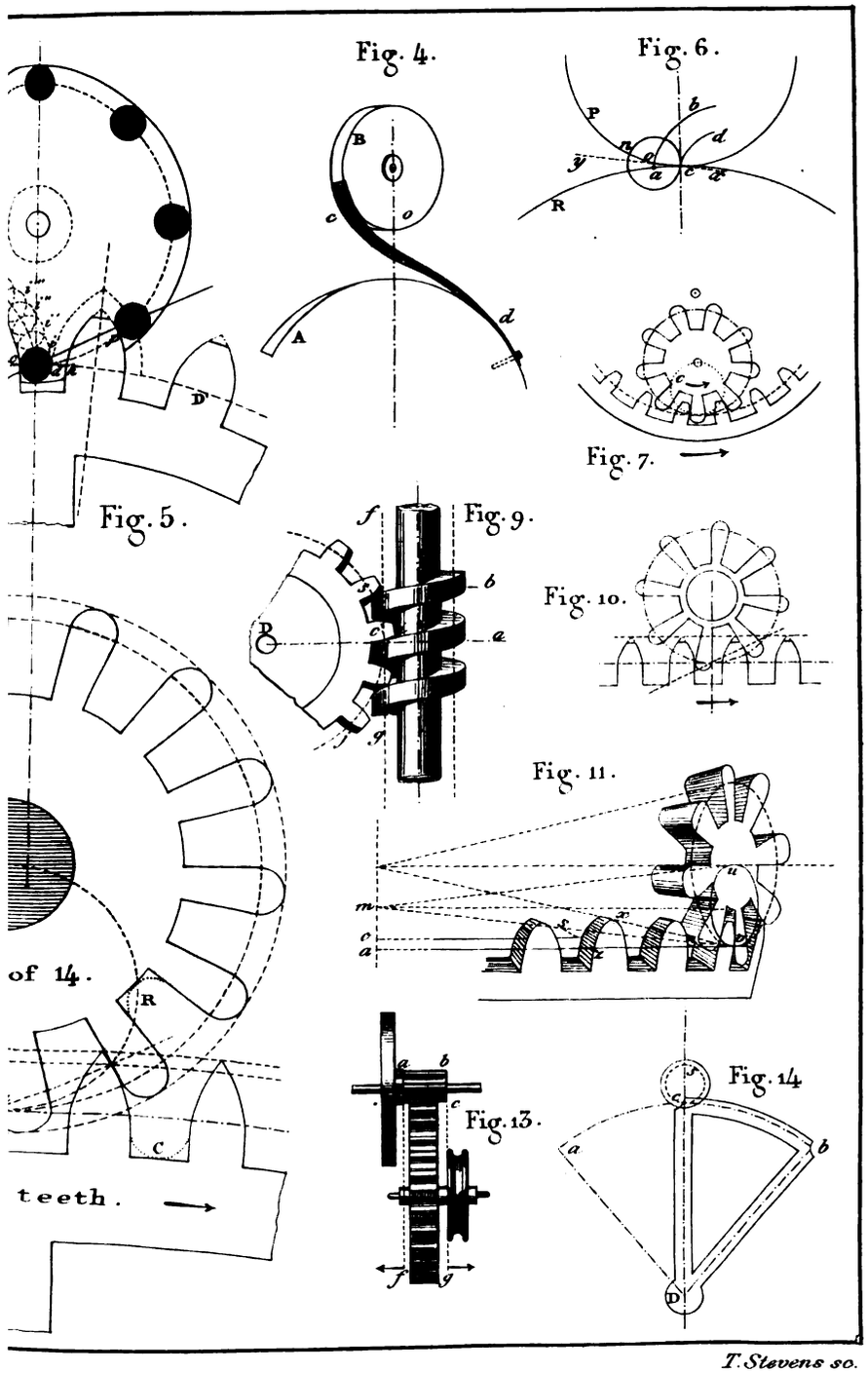
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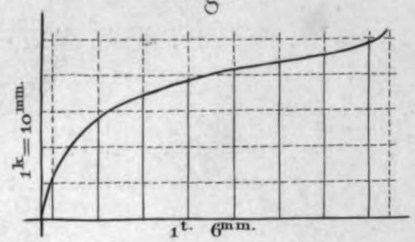

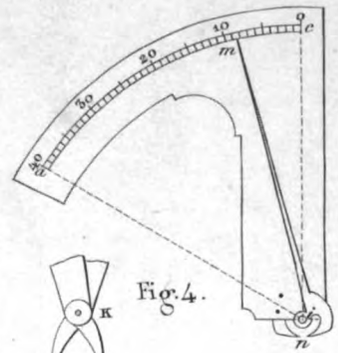

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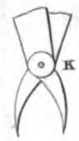

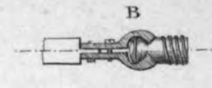

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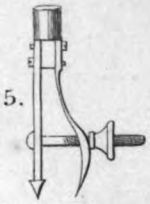

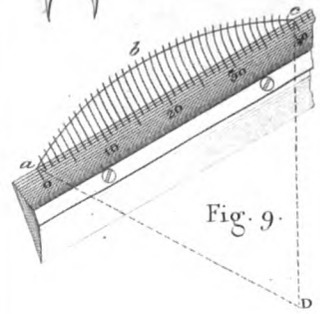

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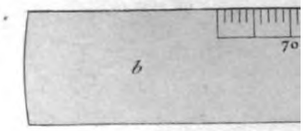

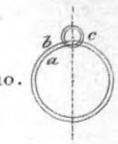

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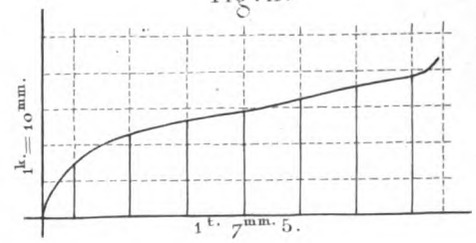

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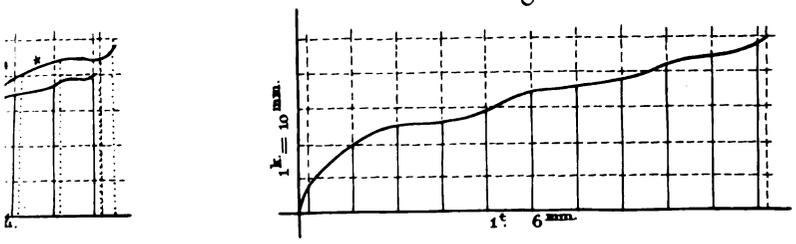

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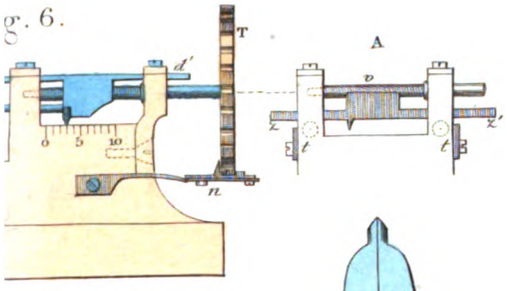

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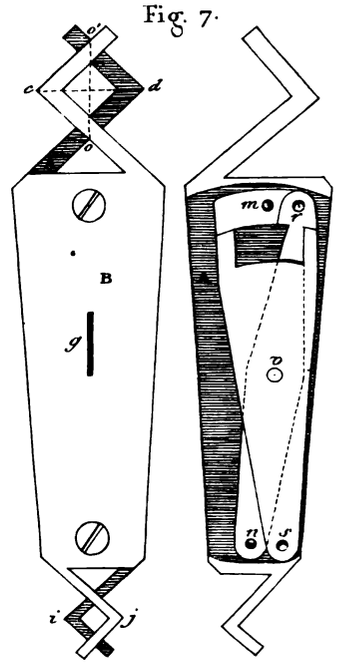

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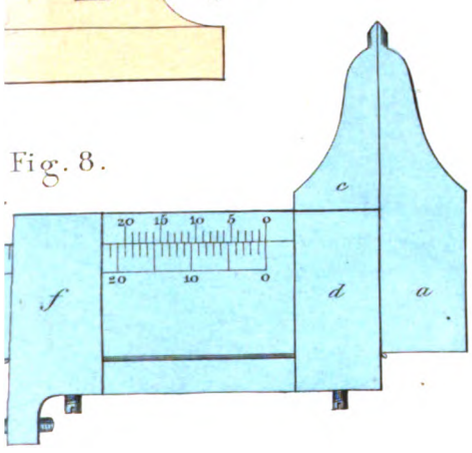

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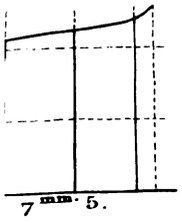

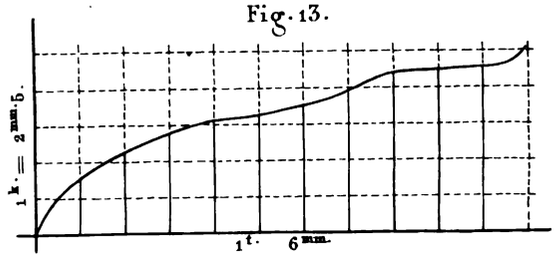

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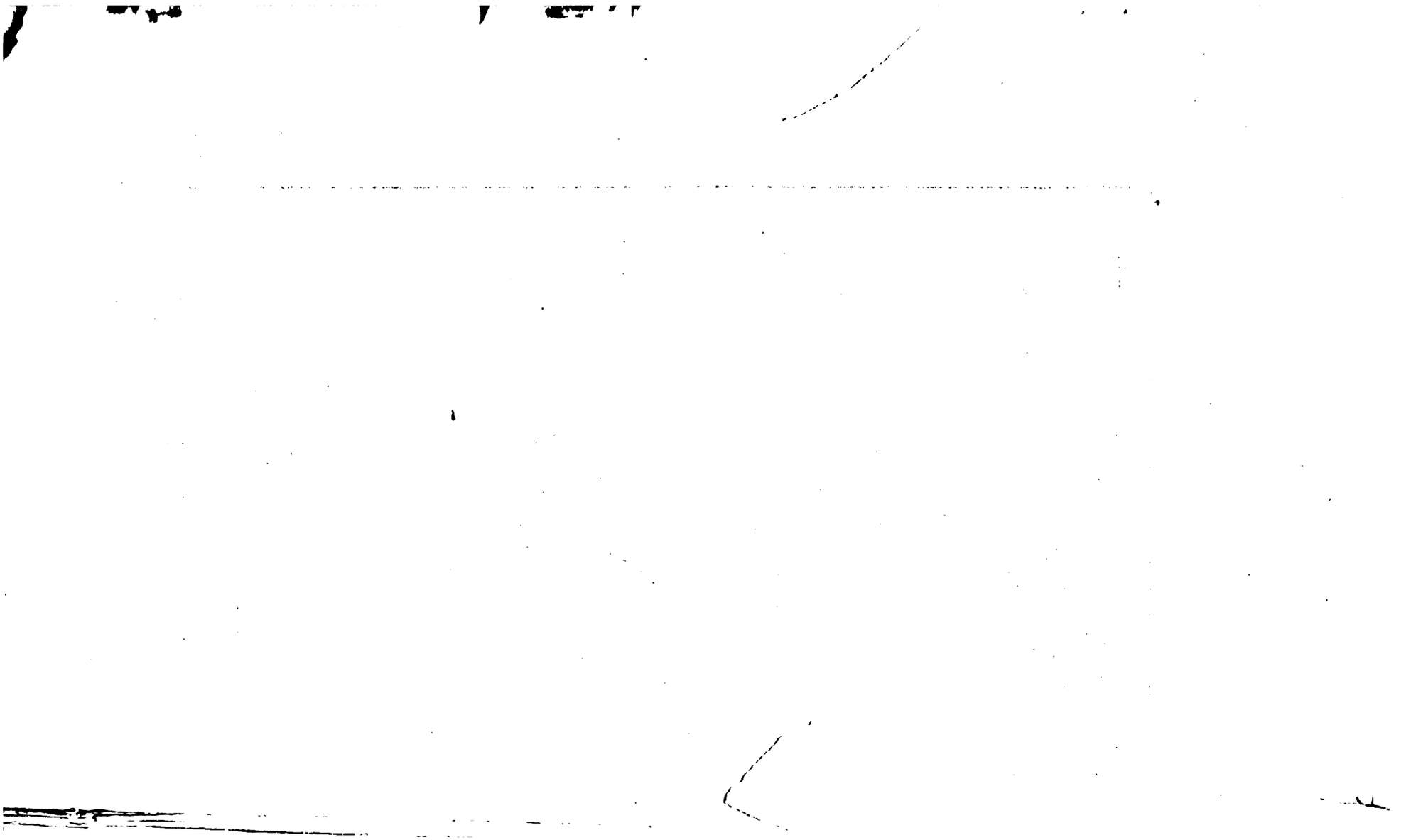

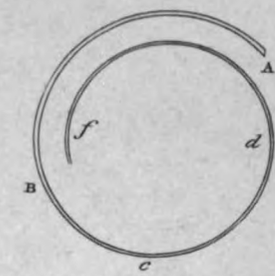

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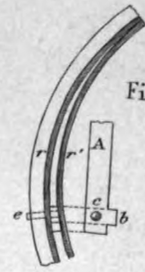

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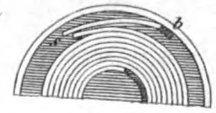

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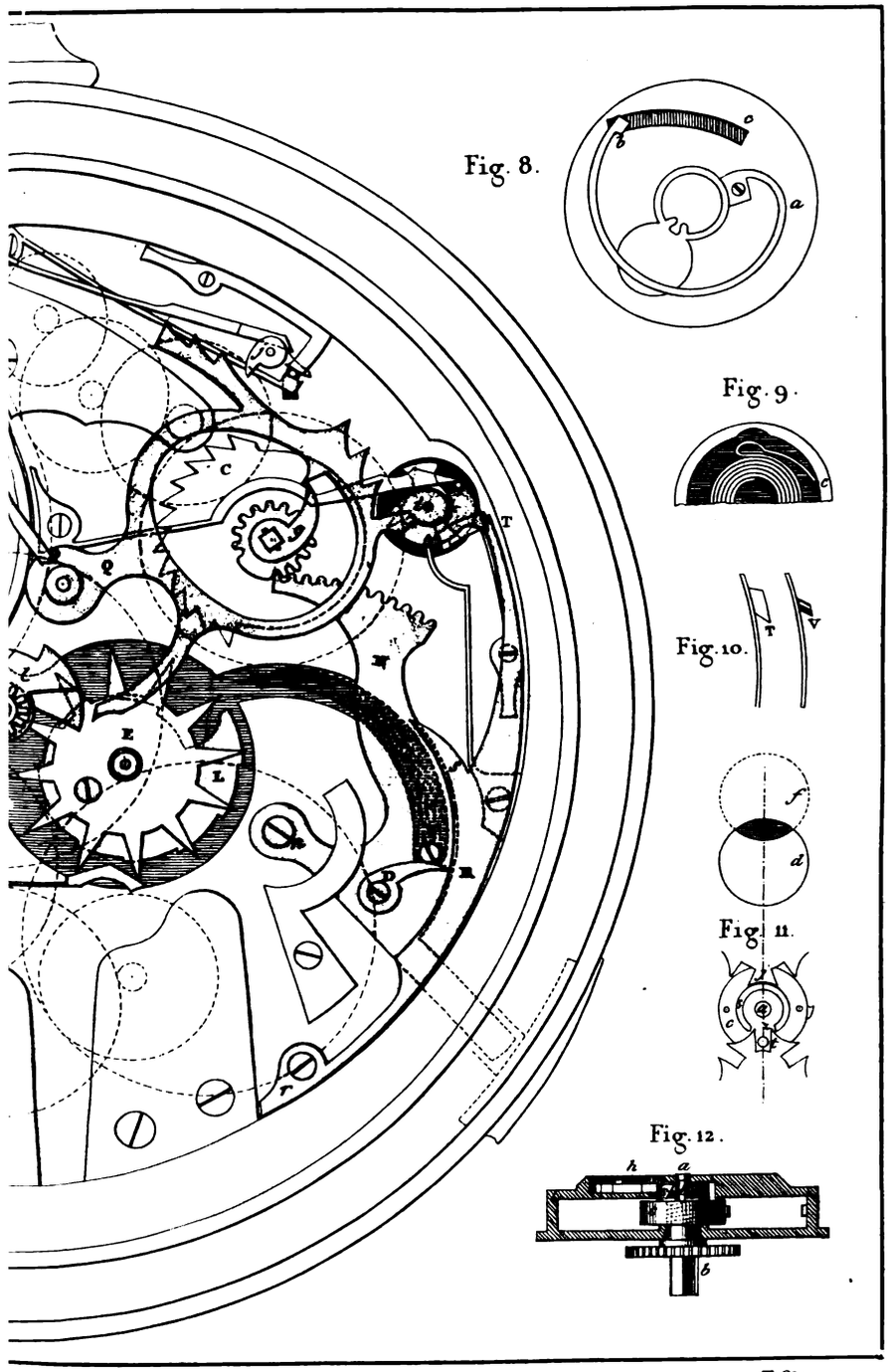

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Fig. 9.

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Fig. 11.

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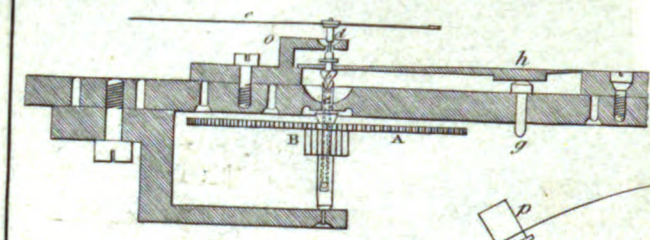

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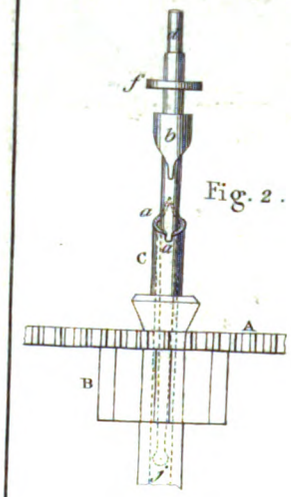

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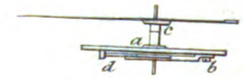

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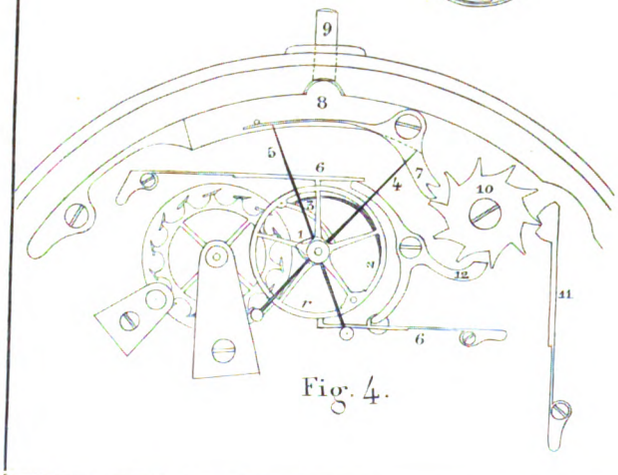

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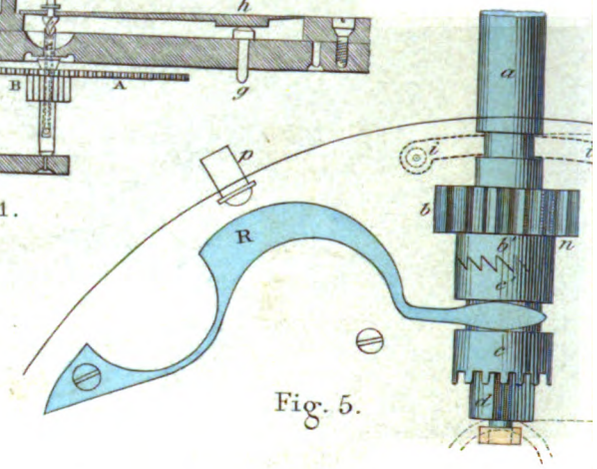

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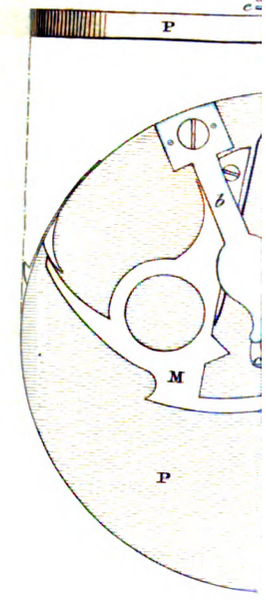

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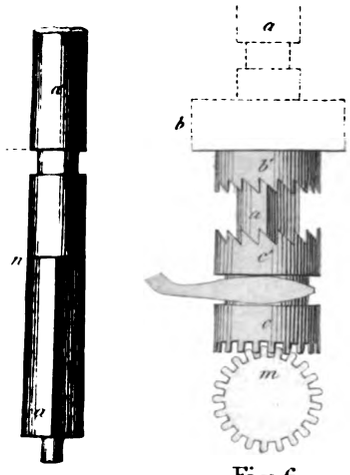

Fig. 6.

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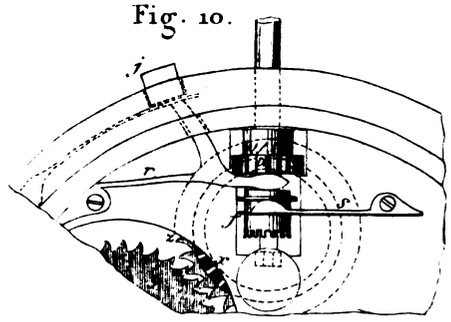

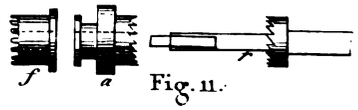

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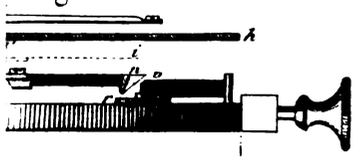

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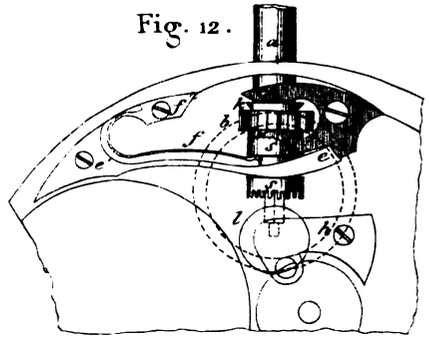

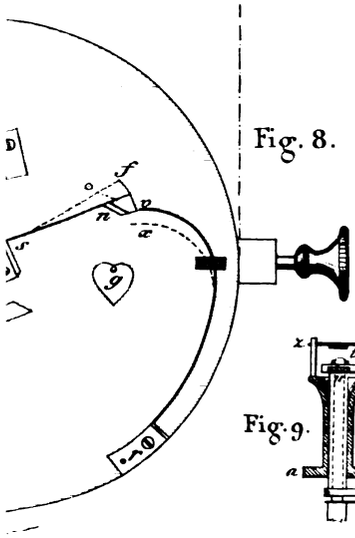

Fig. 8.

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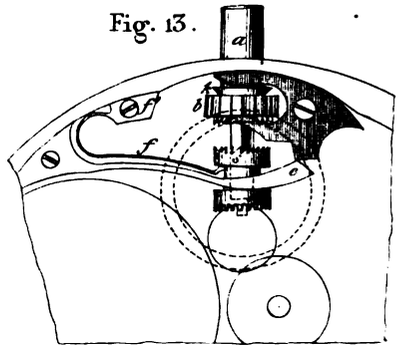

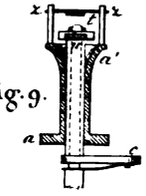

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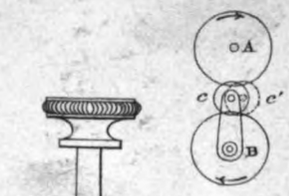

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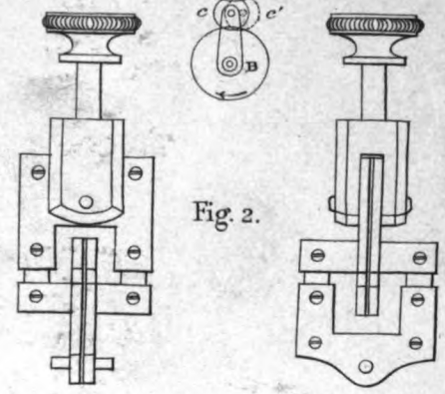

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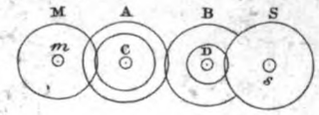

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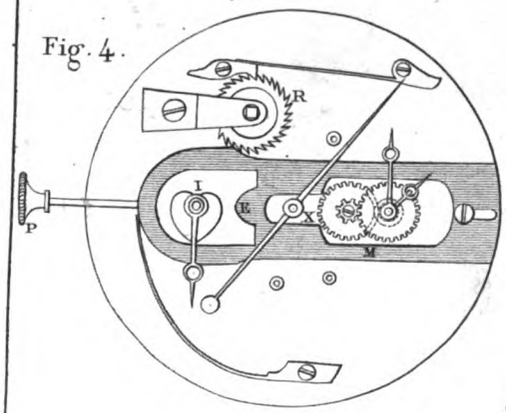

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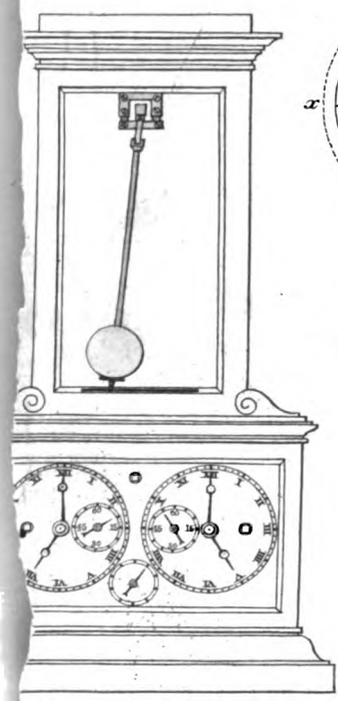

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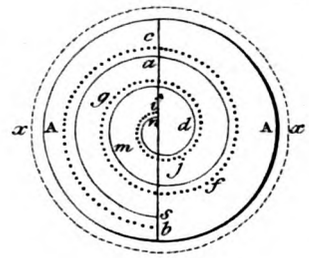

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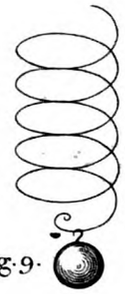

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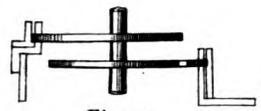

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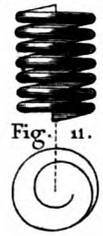

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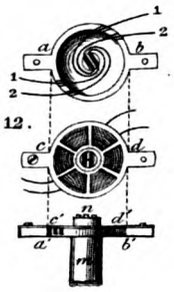

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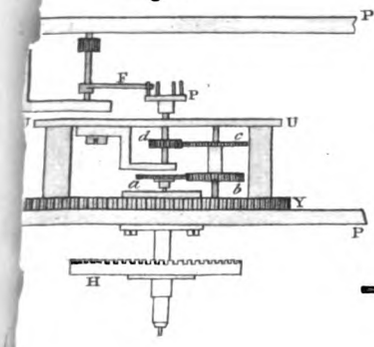

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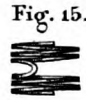

Fig. 16.

Fig. 17.

Fig. 18.

Fig. 19.

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